EXCLUSIVE technical and non-technical articles on design, construction and operation of oil-engines and motorships by the world's foremost writers on marine engineering.

Motorship

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PROFUSELY illustrated with photographic reproductions of the newest designs in international merchant motorship and Diesel-engine construction and auxiliary equipment.

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First D.-A. Engine Tested for Atlantic Liner

Develops 6858 s.hp. on Test at Builders' Works and Is Shipped for Installation in Swedish American Liner Gripsholm

SUCCESSFUL trials have been completed on the first main engine for the Atlantic liner building to the order of the Swedish-American Line, a vessel which will mark the beginning of the era of the double-acting type of Diesel engine.

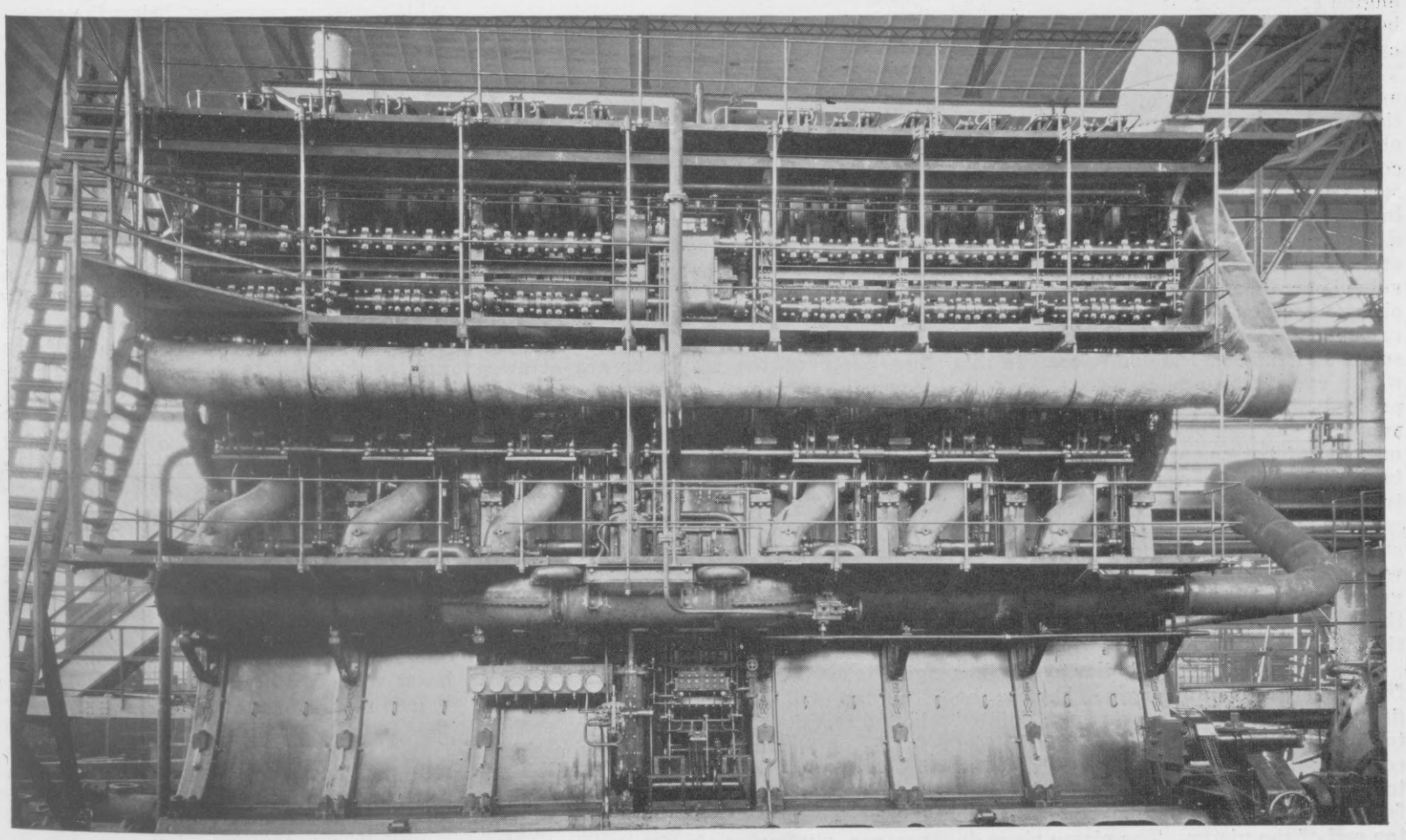
On a 48-hr. test this big engine maintained an average of 6858 s.hp. at just over 124 r.p.m. and showed a fuel consumption equal to 0.38 lb. per s.hp.-hr. Time did not permit an extended series of trials, the owners of the ship being desirous of getting the vessel completed in time for the approaching summer traffic. The shop testing was, therefore, reduced to a minimum, consisting of little more than the 48 hr. run at slightly more than contract power.

GRIPSHOLM, it may be briefly repeated, is to be the first big Atlantic motorliner and is to run between Gothenburg and New York. She will be a twin-screw vessel of about 17,000 tons gross, will have accommodation for about 1650 passengers and will make a speed of around 17 knots.

This speed requires some 13,500 s.hp., which will be developed by two 6-cylinder 4-cycle double-acting engines of the Burmeister & Wain type, with cylinders of 33.07 in. diameter and a piston stroke of 59.05 in. These main engines have no direct-connected compressors, but derive their air from three separate Diesel compressors, each having capacity enough to furnish maneuvering air as well as injection air to one main engine. These compressor sets are driven by engines of 600 b.hp. each. Three additional Diesel engines are installed, driving 330 kw. generators for the electric power and lighting service. All included, the Diesel engines, main and auxiliary, will total over 20,000 indicated horsepower or in excess of 15,000 shaft horsepower.

Special attention has been given to the balancing of all the engines and of the compressors, because this is a passenger ship.

As to the leading details of design, we may quote Mr. Blache, the engineering director of Burmeister & Wain, who read a paper on this subject before the Institution of Engineers & Shipbuilders in Scotland on Feb. 24 last.



Double-acting 6-cylinder engine rated at 6750 s.hp. for the Atlantic motor liner GRIPSHOLM

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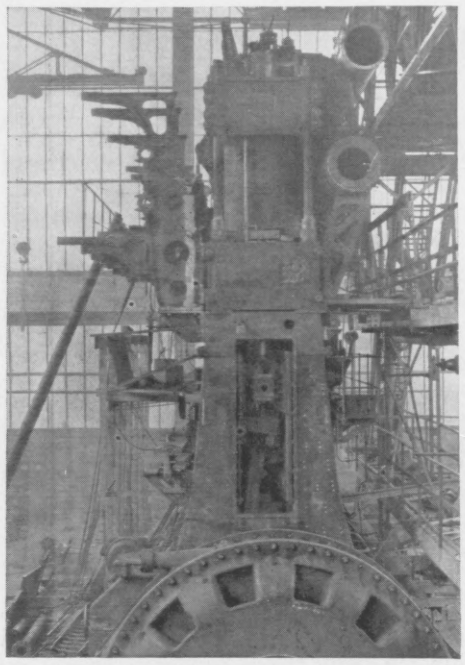
"The difficulties to be overcome in developing the single-acting engine into a double-acting one," he stated, "are encountered in the design of the bottom cover and the combustion in the bottom. The vital part is to give the combustion space around the piston rod a suitable form, to arrange the inlet, exhaust, starting and fuel valves in the cover, as the center part of the same is occupied by the piston rod stuffing-box, and design a piston rod with packing which is able to withstand the high temperatures and pressures of the combustion gases when working. The shape of the combustion space is of most importance, and the volume which is left when the piston is in the bottom position cannot be increased if the clear Diesel principle has to be maintained.

"It is impossible to arrange the valves around the stuffing box in the bottom cover, as the volume would be divided into several pockets in way of each valve, thereby making the combustion space very complicated. This difficulty was eventually got over by arranging a separate combustion chamber for the bottom end, which in a small-size engine gave very promising results, and we subsequently, in collaboration with Harland & Wolff, built an experimental engine developing 1000 hp. in one cylinder. The results obtained were so satisfactory that the design of both 6- and 8-cylinder engines was proceeded with.

"In the present design of B. & W. double-acting 4-cycle engine this difficulty was overcome by arranging the compression volume in one space cast on the side of the cylinder cover. On the down stroke the piston goes right to the bottom of the cylinder, which has a small clearance of almost the same amount as for a steam engine of corresponding dimensions. The bottom cover is hemispherical against the cylinder; the piston is also hemispherical. The large opening for the stuffing box is arranged in the center of the cover, and in this way an exceedingly strong design is obtained. The space at the side of the cover is also designed with comparatively small curved surfaces in view of the high temperatures. The exhaust valve, having a vertical spindle, is fitted at the bottom of the aforementioned space, the inlet valve is arranged at the top in a somewhat oblique position, and the fuel and starting valves are arranged on the inclined end surface of the compression space. In this way a compression space where the compressed air is well gathered around the mouth of the fuel valve is obtained. The mixture of hot compressed air and atomized oil particles flows through the opening from the compression space into the cylinder, filling it during the combustion stroke, while the piston is moving upwards. The distance between the fuel valve and the piston rod is so large that the spray from the fuel valve cannot strike the piston rod direct.

"The cylinder liner is similar to the liner for the single-acting engine, having the flanged end turned downward and jointed to the corresponding flange of the bottom cover by a large number of studs. The top cover is of similar design as the head for the single-acting engine, as also is the top end of the piston. The piston rings do not work further than the top of the cylinder liner, and that part of the cylinder which is cast in one with the top cover is filled up by the top end of the piston. The top cover, which is spigoted into the end of the cylinder liner, is provided with a projecting collar made tight around the liner by means of a rubber ring, so forming an annular space at the top of the liner. This space is provided with a detection hole communicating with the atmosphere in a similar manner as for the single-acting engine. The joint between the flange of the cylinder liner and the bottom cover is also provided with the same safeguard, so that these joints can be controlled for tightness from the outside.



End view of Gripsholm engine

"The engine is provided with through staybolts; they are made in two parts and jointed by means of nuts resting on the surface of the bottom cover. The staybolts pass through the top cover and right through the columns and bedplate, so that the top cover, cylinder liner, bottom cover, columns and bedplate are held firmly together. The bolts are given the initial strain required during the erection of the engine by tightening up the middle nuts at the top surface of the top covers, after first inserting a plate of 1/10 mm. in thickness under the nuts. Thereafter the upper parts of the staybolts are screwed down into the nuts, and the nuts on the top are tightened up so that the middle nuts will be lifted, allowing the 1/10 mm. plate to be withdrawn. This is a very simple method of obtaining correct adjustment of the top cover after overhauling the piston. In this way the cylinder liner is so secured that the elasticity of these parts will be such that any expansion of the liner can be taken up without causing undue stresses.

"The construction of the rest of the engine is similar to the single-acting engine. The top and bottom covers are square, bolted together in groups of three or four for 6- or 8-cylinder engines respectively. The columns are connected horizontally by means of distance pieces forming the top of the crank-casing, where a special scraper stuffingbox is fitted around the piston rod, to prevent the lubricating oil from the crank-casing being conveyed along the piston rod. The top end of the connecting rod is of the ordinary forked type. The piston rod is secured by means of a nut at the top and bottom of the square part of the crosshead; the guide shoes are fitted on the end of the crosshead pins, and work on loose guide-plates bolted to the columns between two cylinders. These guide shoes are forced lubricated, and for which no separate water cooling is arranged. The piston rod is made of steel, having a flange at the top end tapered to the rod. The piston is in halves, and secured around the flange at the end of the piston rod; a cylindrical cast-iron liner of the same material as used for the cylinder liner is passed through the bottom half of the piston. The piston-rod liner is jointed to the inside flange of the bottom half of the piston, and centered at the lower end around the piston rod. The piston rod stuffing-box consists of a separate chamber which is water cooled separately, fitted in the bottom cover, and provided with ten self-expanding cast-iron piston rings. The rings resemble the piston rings for an ordinary Diesel engine, but are cut in such a way that they can expand toward the rod.

"The piston and piston rod are cooled by means of forced lubricating oil which passes through a single telescopic pipe fitted to the end of the crosshead pin, and led up between each two cylinders in a similar manner as for the single-acting engines.

"The camshaft for the valve gear is arranged about midway between the top and the bottom cover. It is provided with cams for the valve gear for the top and bottom covers and fitted in bearings together with the two reversing shafts which, during reversing, bring the rollers free of the cams. These bearings are arranged on the columns in such a way that one cylinder with the top and the bottom cover can be removed without dismounting the camshaft, which is operated from the crankshaft by means of a twin-roller chain drive, each strong enough to transmit the whole force from the crankshaft to the camshaft. The camshaft drive is fitted in a separate inclosed casing standing free in the center of the engine, in a similar manner to that for the single-acting engines. It has, therefore, the same elasticity to follow deformations of the hull of the ship. The fuel pumps are arranged on the above-mentioned casing in two groups for the top and bottom respectively. and the two maneuvering handles operate the fuel pumps for the top and bottom respectively. The reversing and maneuvering valve-gear and valves are designed similar to those for the single-acting engines. The exhaust valves are not complicated by any extra water cooling, as the temperature of the exhaust gases of a 4-cycle engine is not so high as to be injurious in any way to an exhaust valve having seats of special alloy."

It is reported that both main engines for the GRIPSHOLM have now been shipped from Copenhagen to the builders of the vessel in England. If expectations are realized the GRIPSHOLM will arrive in New York on her maiden voyage during the month of June.

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Another 600 hp. Motor Tug with Direct Drive

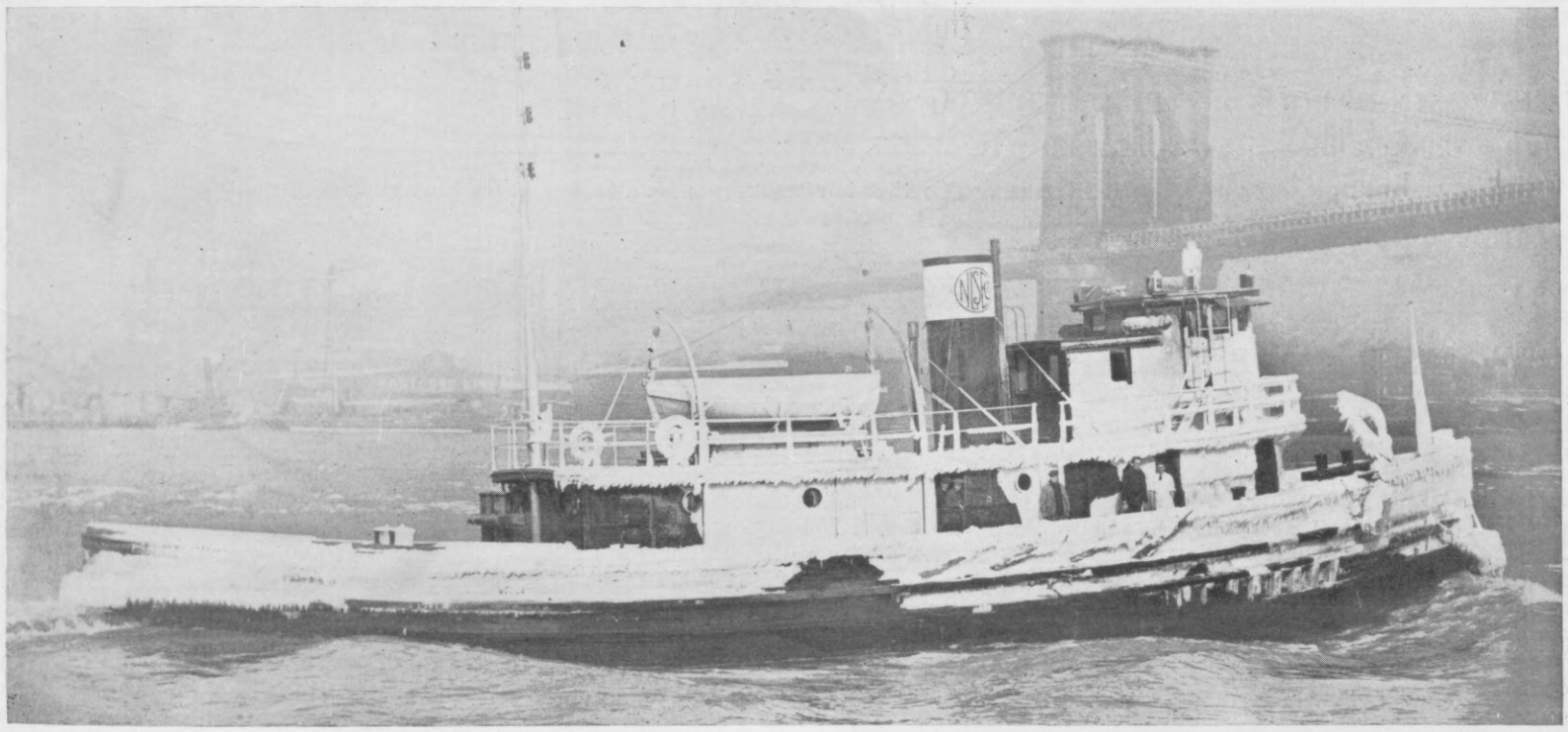
"Lion," a Towboat of 218 Tons Gross for Heavy Work Is Commissioned in the Port of New York

LARGELY as a result of the success of the motor tug JUMBO, which was put into service about this time last year, the New London Ship & Engine Company has gone ahead with a second towboat of about the same size and power.

LION, as the new boat has been named, is registered as 218 tons gross, with a length of 95.9 ft. between perps., 25 ft. inside breadth and 13.8 ft. deep between keel and deck. Her main engine is a 6-cylinder Nelseco set of 600 s.hp. with cylinders of 16½ in. diameter and 24 in. stroke.

An uncommon feature of the installation is that when the main engine is operating all the auxiliary units are driven by it. This refers not only to the circulating water pump, lubricating oil pump, fire and bilge pump and fuel oil transfer pump, but comprises also a 110-volt generator. The problem of utilizing a generator driven at variable speeds was long ago worked out for railroad and automobile purposes. It is linked up with the use of a storage battery, and arrangements are provided for automatically cutting out the generator when the engine speed is too low or when the engine is reversed. In the tug, LION, this generator is driven from the propeller shaft.

When the main engine is not under way, the electric power and light is furnished by a 5 kw. generator direct-connected to a 9 hp. single-cylinder Venn-Severin oil engine. The auxiliary fire and bilge pump, the auxiliary circulating water pump, the auxiliary forced lubrication pump and the De Laval oil purifier are each separately driven by electric motors. On deck there is an electric capstan forward and aft there is an electric steering gear. For heating there is a coal-burning Arcola furnace.



LION was put into service when the winter weather was at its worst

Bids on 54 Auxiliary Engines Invited by U. S. S. B.

For Use on the 18 Steamers to Be Converted from Steam to Diesel Power —Generating Sets of 75 kw. Will Be Installed

SPECIFICATIONS have just been issued by the Shipping Board on the Diesel driven generator sets of 75 kw. that are going to be installed on the 18 vessels to be converted into motorships. Three of these sets will be placed in each ship, and they will be of the full Diesel type with either two or three cylinders and may be either 2-cycle or 4-cycle. A preference is expressed for the use of air injection. They are to be mounted on the same base as the 240-volt generators which are to be of the d.c. type.

Engines are required to be of the fully inclosed type, fitted with forced feed lubrication. The rating must be such that the engines can continuously develop their b.hp. at the stated r.p.m. under normal service conditions and be capable of taking 25 per cent overload for ½ hour. The speed must not exceed 300 r.p.m. and should preferably be about 250 r.p.m. A higher piston speed than 800 ft. per min. is ruled out, and the mean indicated pressure (referred to i.hp.) must not exceed 90 lb. per sq. in. in 4-cycle engines or 85 lb. per sq. in. in 2-cycle engines.

The same fuel oil is to be burnt by the auxiliary engines as will be supplied for the main engine, and the auxiliary sets are required to be able to run satisfactorily on this fuel with a clear exhaust for periods of 600-700 hours with loads ranging from full to 60 per cent. The exhaust must also be clear when the engines are operated for 10 and 24 hours at quarter load.

Engines are to be self-contained with attached fuel pumps, forced lubrication pumps, injection air compressors and scavenging pumps where required.

Alternative proposals have to be submitted for an engine with an over-sized compressor to act as a standby to the main engine in case of failure of the main compressor. Other alternative proposals are to be submitted for supplying a self-contained engine with an air-compressor driven from the extreme outer end of the generator shaft by means of a magnetic type clutch.

Shop tests of more than usual severity are required. The first engine must make a 15 days' non-stop run at rated power and r.p.m., followed by 24 hours at 10 per cent overload and one-half hour at 25 per cent overload. Other engines must run 12 hours at full load and 2 hours at 25 per cent overload. Sea trials of 12 hours duration must also be met.

More details about these engines will be found in the March OIL ENGINE POWER.

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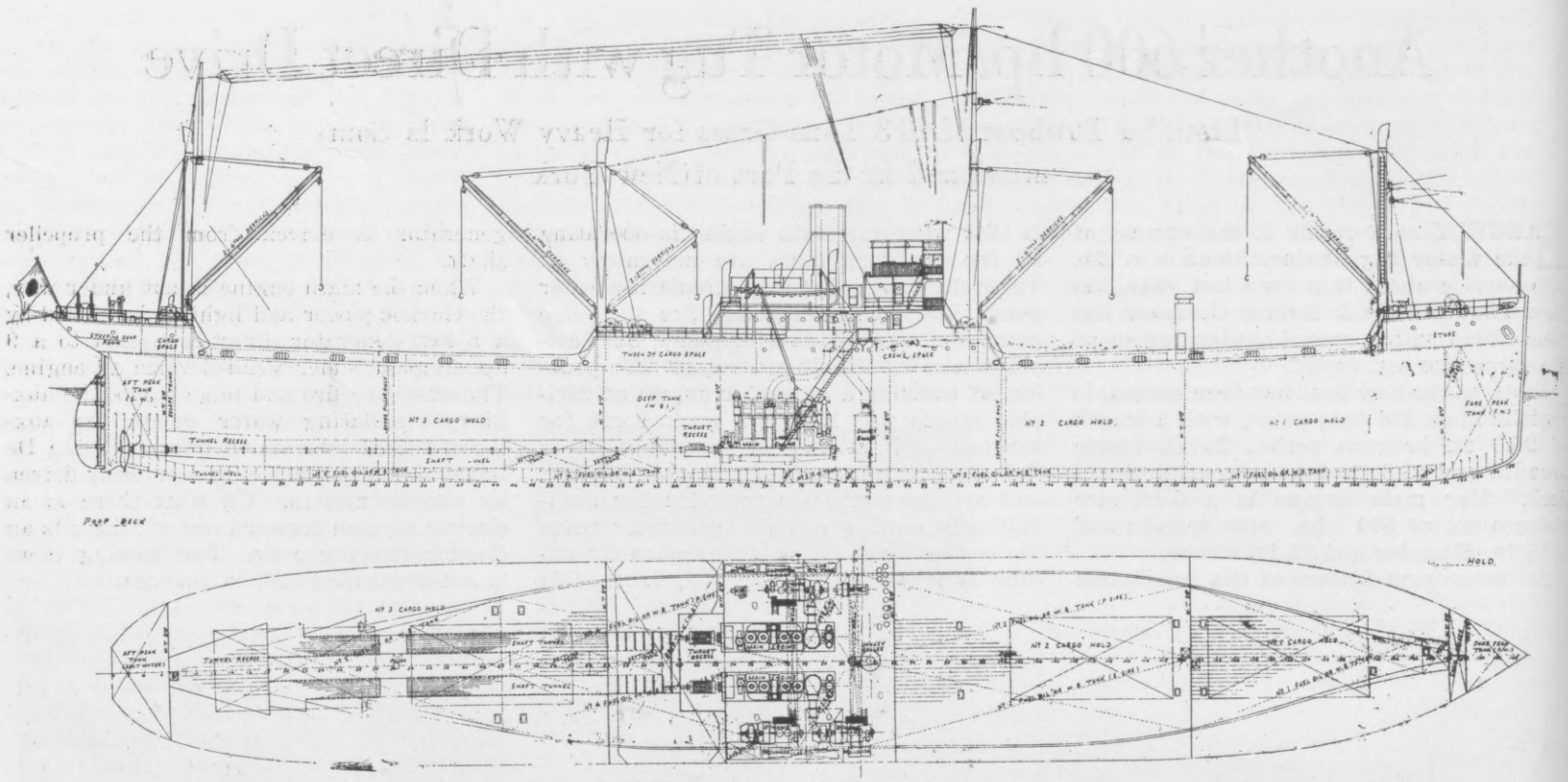


Profile and plan of the lumber motorship that is attracting much attention at the lumber ports of the West

Improved Lumber Carrier Is Diesel-Engined

Pacific Coast Interested in New Motorship with Large Clear Holds and Unobstructed Decks for Lumber

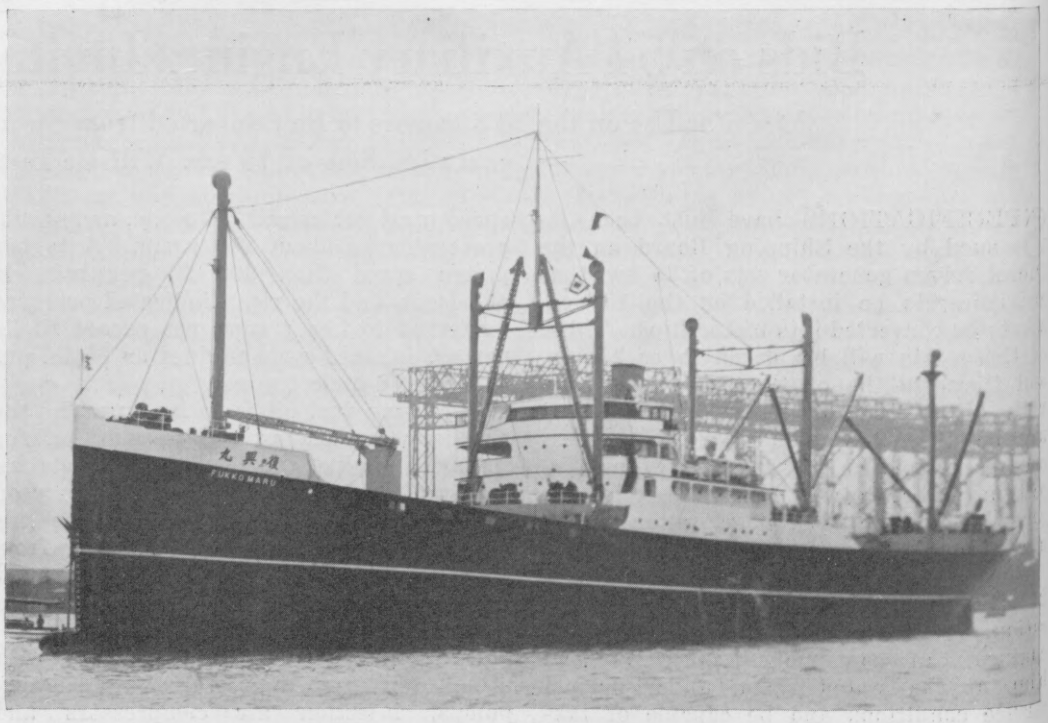
SOMETHING new in lumbership construction is exemplified in the motor vessel FUKKO MARU, scheduled on the Pacific Coast after her maiden passage from Yokohama. New features of marine architecture and new theories of shipping economics are embodied in this vessel. What has focussed Pacific Coast attention on her is not merely that her holds and decks are quite free and unobstructed, but also that she represents the highest priced kind of tonnage and yet is intended for the lowest rate cargoes.

She measures about 3600 tons gross, being 350 ft. long, 50 ft. beam, and 27 ft. deep, with a deadweight capacity of some 6000 tons. Except for the center pillar between Nos. 3 and 4 holds, there is no break whatever in the after cargo space. Forward there is a bulkhead between Nos. 1 and 2 holds, and there is also a center pillar in No. 2 hold, but otherwise this cargo space also is unobstructed. The deck is entirely clear between bridge and poop, the masts being stepped in the superstructure and the winches being mounted on the houses. The same is true of the forward deck space except that a mooring post for the deck-load is located over the bulkhead between Nos. 1 and 2 holds. The masts in themselves are a feature, being of sufficient diameter to act as ventilators for the entire hold space. It is said that the design of this vessel raised so many debatable points that her classification by Lloyd's was long in doubt, but she has actually been classed 100 A1.

Driven by two Sulzer 4-cylinder engines of 800 s.hp. each, she makes about 240 miles per day on a consumption of about 7½ tons. With a fuel tank capacity of 740 tons in her double bottom she can steam between 90 and 100 days without rebunkering. There is additional capacity of 855 tons in the deep tank, which can be used for oil cargo, but is chiefly for ballast.

There are three generator sets driven by 2-cylinder oil engines and providing power for all the engine-room auxiliaries and deck machinery. The latter consists of an electric windlass, eight 4-ton winches and two 3-ton winches. The two smaller winches serve the hatches of the tween deck cargo space and deep tank, while the larger units are for the extensive hatches over the main holds, these hatches measuring 39 ft. 9 in. by 20 ft. and extending, as can be seen, the full length between the breaks of the deck-housings.

This improved type of lumber carrier, designed for use in the Japanese coastwise trade and occasional trips to Pacific ports, is owned by Suzuki & Co., a very progressive firm of Japanese merchants.



M. S. FUKKO MARU at the yard of the Kobe Steel Works, Japan

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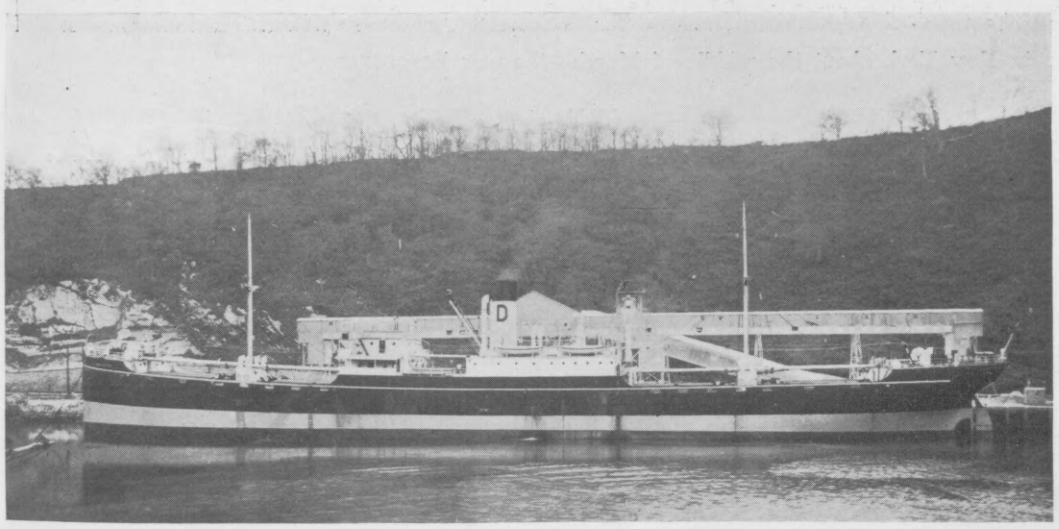
Motorships Nosing Into the Tramp Business

First of the Motor Vessels in the Dalgleish Fleet of 31 Tramps Arrives in Pacific Ports with Miscellaneous Cargo

ELMWORTH, a single deck vessel of the familiar three-island type, is the initial venture of what may develop into the replacement by motorships of the entire fleet of 30 tramp steamers owned by R. W. Dalgleish, Ltd., of Newcastle-on-Tyne. A sister ship, OAKWORTH, will soon be in service.

ELMWORTH is on a proper tramp voyage, picking up cargoes wherever they are to be had. She sailed on her maiden passage Dec. 9 bound for the Pacific Coast with a full cargo of bricks, china, clay, pig iron and coke, loaded at Channel ports. This was discharged at San Pedro, San Francisco and Vancouver, B. C. Then she started loading lumber at Vancouver and completed at Portland, Ore., for Sydney, Australia.



First of a big tramp fleet, the ELMWORTH

She was brought out by Capt. T. H. Wilson, and on the trip across the Atlantic encountered terrible weather, taking just under 24 days for a non-stop run of 5047 miles to the Panama Canal. Her average speed was only 8.4 knots on account of having been hove to several times in gales.

She made a very quick transit of the Canal, entering the sea channel on the Atlantic side at noon and bidding farewell to the channel out to sea on the Pacific side at 8 p. m. Up the Pacific Coast her best averages were 10½ knots. After her calls at San Pedro and San Francisco she reached Vancouver on Feb. 8.

Most of the way out her engines were run at 90 r.p.m. On her trial trip she made 10.9 knots at 92 r.p.m. Subsequently on a run from Fowey to Antwerp the log showed an average speed of 11.1 knots at 90.8 engine revolutions per minute.

ELMWORTH has a length overall of 400 ft., beam 53 ft., molded depth of 29 ft. 7 in. and loaded displacement of about 12,000 tons with 8000 tons deadweight. A Harland & Wolff engine with six cylinders of 29.13 in. diameter and 59.06 in. stroke propels the ship. This long stroke type of engine, specially designed for freight carriers, develops about 1850 s.hp. at 90 r.p.m. The engine-room staff consists of seven engineers and assistants, two electricians and a motorman.

All the auxiliary machinery and deck machinery is electrically operated, power being supplied by three 2-cylinder 100 b.hp. B. & W. type Diesels direct-connected to generators. One of these units is sufficient to supply all electric power needed at sea, and two are used in port with all the winches operating; the third engine is a stand-by. A motor-driven auxiliary air compressor is installed in addition to the 3-stage air-compressor driven off the main engine—while the fuel transfer pumps, lubricating oil pumps, circulating water, bilge, and ballast pumps are all driven by electric motors. There is a small oil-fired boiler in the engine room for steaming out tanks, but steam is not used for any other purpose. The heating throughout the ship is by means of electro-vapor radiators.

Fuel before going into the service tank is passed through a De Laval separator with a capacity of 300 gal. per hr. This centrifuging removes the water, grit and other damaging substances. Chief Engineer James Moss stated in Vancouver that after 40 days' coasting around the U. K. and crossing the Atlantic he removed the valves of the main engines at San Pedro—as there was nothing else needed attention—and found them in such splendid condition that they would have continued to function properly for the voyage on to Australia without any attention. This he attributed to the centrifuging of the fuel oil.

His experience in other motorships with dirty oil was that after such a trip the valves would need regrinding or even turning in the lathe to smooth off the burned surface where the hot gases had attacked the pitting caused by impurities in the fuel. The fuel consumption of the ELMWORTH is 7.65 tons a day for all purposes when making 10.5 knots, and the oil used on the trip out was a British-Mexican Diesel oil 26 deg. Beaumé and 0.895 gravity.

On account of the double-bottom fuel tanks being new and not having been particularly well cleaned out, a certain amount of dirt from this source got mixed with the fuel oil, and it was found necessary at first to clean out the centrifuge every couple of days. As the fuel was drawn from the tanks, however, they got cleaner, and the refuse from the last 900 gal. of fuel run through the separator before Vancouver was reached amounted to about a pailful of water and muck. Additional fuel had been loaded in California. The fuel capacity is 924 tons, all in double bottom tanks, giving the ship a cruising radius of 30,000 miles at full speed, a very useful feature in a tramp. The fuel consumption of the ELMWORTH averages 7.6 tons a day for all purposes when making 10.5 knots.

Mr. Moss states that the lubricating oil consumption works out at 3 gal. a day for cylinder oil for the main and auxiliary Diesels, 1½ gal. for the compressor and about 1½ gal. for propeller shaft, steering gear and other outside machinery.

This vessel is equipped with Clarke-Chapman electric winches with worm drive. They are silent running machines, and each winch is a self contained unit. The ship is also fitted with electric windlasses, and electro-hydraulic steering gear with telemotor control from the bridge.

An examination for cartographic engineers in the Coast & Geodetic Survey will be held by the U. S. Civil Service Commission in the spring. These positions carry an entrance salary of $1,860 a year, and pay is advanced without change in assignment up to $2,400 a year. Applications will close March 28.

To fill vacancies in the Steamboat Inspection Service at entrance salaries ranging from $2,700 to $4,200 a year, the U. S. Civil Service Commission will hold an open competitive examination for positions of local and assistant inspectors of hulls. The vacancies are in ports of the Great Lakes and on the Central Western rivers. Applications must be filed before March 28.

Resumption of shipbuilding in the Leningrad district has had an influence in reviving the construction of Diesel engines at the plant now known as the Russian Diesel, which formerly was operated under the title of L. Nobel & Co. There are at present under construction in those works four engines of 750 s.hp. each for oil tankers, and two engines of 1900 s.hp. each for refrigerator ships. This is quoted on the authority of a Moscow magazine.

Gunnar Knudsen of Porsgrund, owner of the new motorship BORGESTAD, built by Burmeister & Wain, was previously Prime Minister of Norway. He has another vessel of the same type and size on order.

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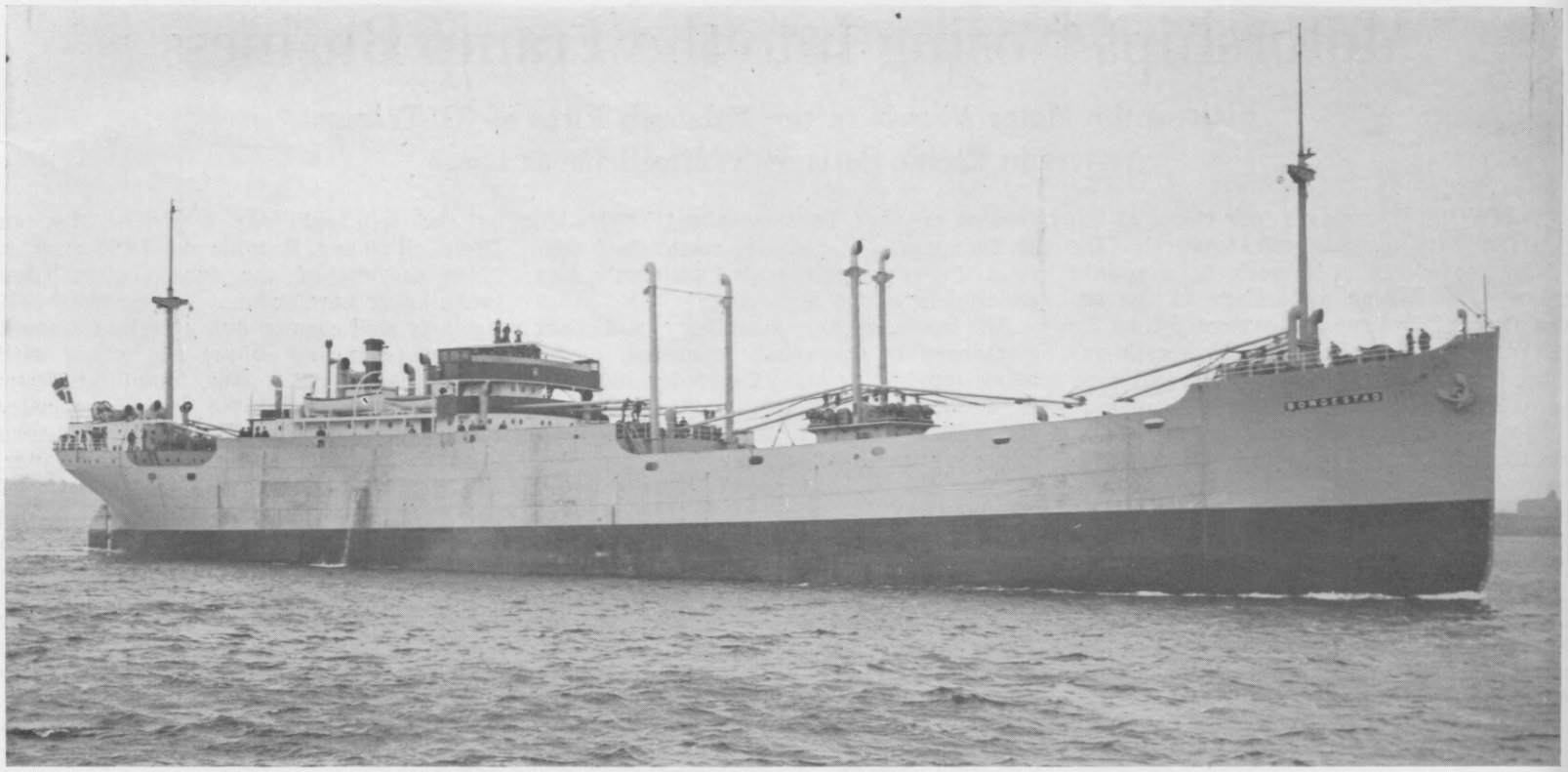

Expected to arrive in New York about the middle of March—the new BORGESTAD

New Motorship Service Between New York and S. A.

Four Vessels of About 7000 Tons d.w. Each Are to Be Put on the South American Route by Garcia & Diaz of New York

A NEW motorship service between New York and South American ports is being established by the Linea Sud-Americana, the sole managing agents of which are Garcia & Diaz of New York. This service will at the start keep four vessels in operation. Pending the completion of three vessels now building at the Burmeister & Wain yard for the account of the Linea Sud-Americana the service is being initiated by two chartered vessels, the VINLAND and BORGESTAD. The former was completed about three months ago and sailed from New York on its first run in this service during the month of February. The BORGESTAD has just come out of the B. & W. yards and is expected to load in New York about the middle of March. She has been chartered for two trips only.

The three vessels building for Garcia & Diaz will be named respectively PRIMERO, SECUNDO and TRESCERO. These three ships

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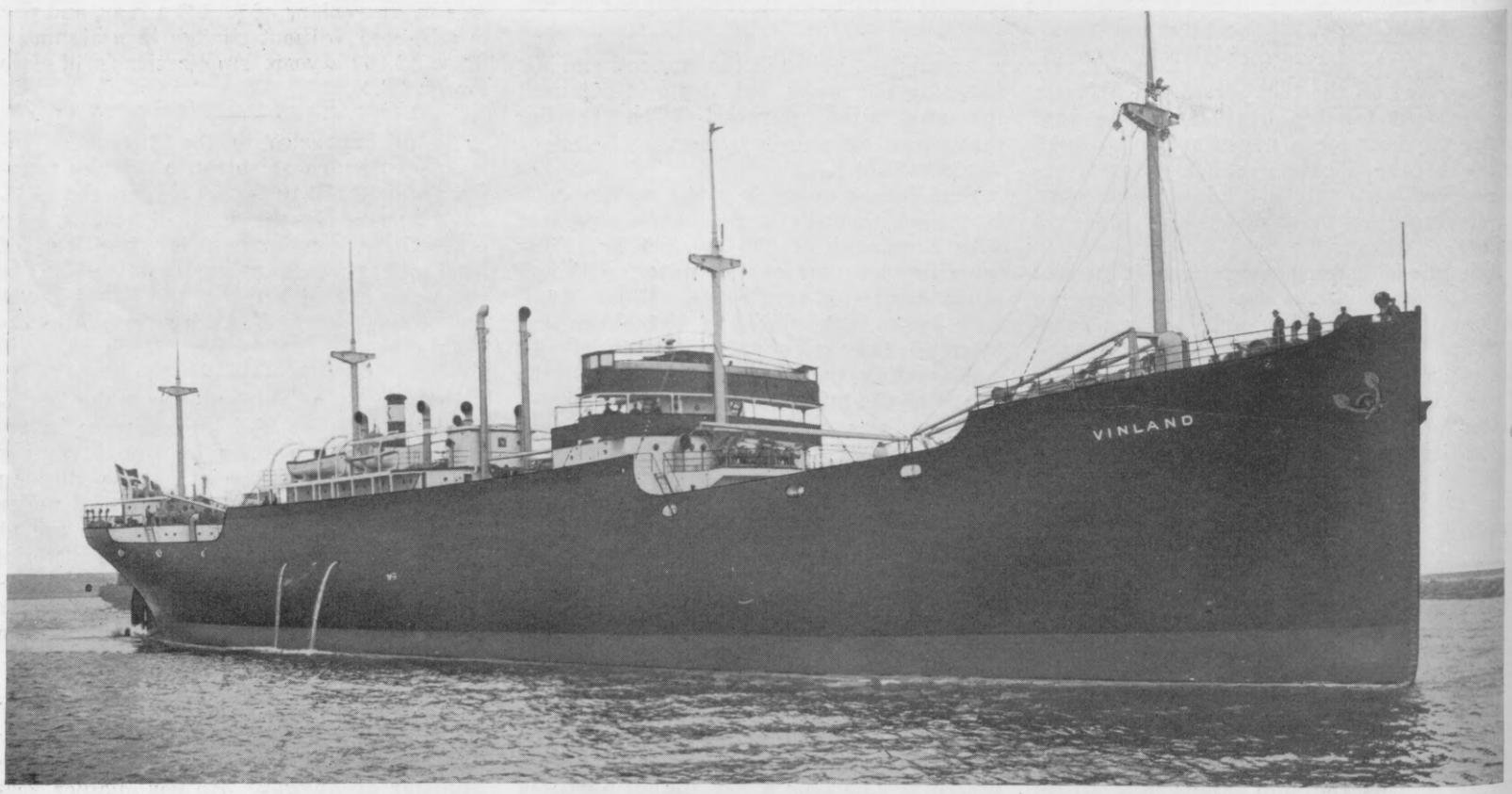

Chartered for the new motorship service between New York and South American ports

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Grace Line Joins Ranks of Motor Tug Owners

A 320 hp. Towboat Added to the New York Floating Equipment of the W. R. Grace & Co. Lines

GRACE is the name of the new motor tug which W. R. Grace & Co. recently put into commission for their own general needs in the Port of New York. The Grace Line operates a fleet of combination freight and passenger vessels between that port and the Pacific Coast of South America via the Panama Canal. They require tug services for their vessels in docking or sailing, and the handling or towing of lighters for loading or discharging at their piers means also a lot of work.

GRACE was ordered for this service. She was built in Baltimore by the Spedden Shipbuilding Co. and has an Ingersoll-Rand engine. Measuring 76 ft. 6 in. b.p., 19 ft. beam and 9 ft. 9 in. molded depth, she makes a maximum of better than 11 knots with her 320 hp. engine. This engine, which has 6 cylinders of 13 in. diameter and 19 in. stroke, develops its rated power at 250 r.p.m., and, like all Ingersoll-Rand engines, is of the airless-injection type.

Electric power and lighting is provided by an 8 kw. generator, driven by a 12 hp. Bates & Edmonds oil engine, which also can be clutched to an auxiliary air-compressor. A second oil engine of the same power is direct-connected with an Ingersoll-Rand air compressor to supply maneuvering air. For fire and bilge service there is a Blackmer rotary pump of 100 gal. per min.

An unusual type of steering engine is fitted. It is a simple reversible air motor, controlled by a lever which also opens and closes the compressed air throttle. It gives much quicker response than is possible with a handwheel, and during the two months' trial it has had in everyday work has proved entirely satisfactory. A device of this sort is quite a boon to a harbor towboat and costs little to operate on a motortug.

New tug operated by W. R. Grace & Co. for their own pier service in New York

New Motorship Service between New York and S. A.

(*Continued from previous page*)

will be of about the same size as the VINLAND and BORGESTAD; that is to say, averaging about 7000 tons d.w. capacity. They will be of the twin screw type with the new design of trunk-piston engine developed by Burmeister & Wain.

VINLAND, one of the latest ships equipped with the Burmeister & Wain standard light-built type of engine, was completed at the end of 1924. She is 367 ft. long b.p., 53 ft. 6 in. molded breadth and 27 ft. 6 in. depth, with a d.w. capacity of about 7500 tons, and has an average sea speed of 10½ knots. Her two engines are rated at a total of 2500 i.hp. when turning at 150 r.p.m. Her owner is Knut Knutsen of Haugesund.

BORGESTAD is the latest B. & W. motor vessel, and is of 6800 tons d.w. capacity, measures 350 ft. b.p. with 51 ft. 3 in. molded breadth and 27 ft. 3 in. depth molded to the upper deck. Her registered gross tonnage is 3924 tons. An unusual feature is the large fuel capacity, which is sufficient for a steaming distance of about 50,000 miles.

The propelling machinery consists of two 6-cylinder B. & W. trunk-piston type engines of 1700 s.hp. at 145 r.p.m. All the auxiliary machinery throughout the ship is electrically driven, current being supplied by three dynamos, two of 33 kw. and one of 66 kw., direct-connected respectively to two single-cylinder and one twin-cylinder Diesel engines of B. & W. construction.

Engine room machinery and shafting, together with donkeyboiler, weigh no more than 307 tons, which is equal to only 3.3 per cent of the total displacement of the ship when fully loaded. The fuel consumption at 154 r.p.m. was measured on the trials to be 0.362 lb. per s.hp. hour inclusive of the fuel used by the auxiliary engines. This vessel is owned by the A. S. Borgestad of Porsgrund, Norway, a company controlled by Gunnar Knudsen of the same city.

Garcia & Diaz have a strongly established position in the trade between New York and Spain. They have also been operating vessels for a considerable period between Spain and the River Plate. This is the first time they have entered the trade between North and South America. With the advent of the new motorship service this trade route may be regarded as one of the most competitive motorship routes in the world. Some of the Wilhelmsen motor vessels are operating regularly on this run, and Lamport & Holt have put some of their best motorships into the same service. This will make it hard for the steamers.

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Further Considerations on the Rotorship

Practical Experience Is Regarded as the Essential Key to a Proper Appraisal of the Value of the Flettner System of Propulsion

WHILE the BUCKAU is on her maiden voyage and while the inventor, Anton Flettner, and his representatives in various countries are trying to live down the exaggerated reports first published about the capabilities of the rotorship, it is timely to make an estimate of the value of this form of vessel.

A statement in the *Frankfurter Zeitung*, which is generally regarded as the most responsible commercial newspaper in Germany, sounded the first warning note in Germany against undue optimism. It appeared as a contribution of the Hamburg correspondent of that journal and seems to express what may perhaps be taken as the opinion of the Hamburg-America Line.

"Since the sensational discovery of Herr Flettner became generally known," says the statement, "reports have been spread both in foreign, as also in German, newspapers according to which German shipowners were already considering extensive building programmes on the basis of the Flettner rotor. Among other things, it was even declared that the Hamburg-America Line, which is financially interested in the Flettner Company, had given an order for the building of the 10,000 ton Flettner ships. In order that false ideas may not be spread abroad, it seems to be only right to give a warning against exaggerated expectations. The real state of affairs is that any sort of views as to building are merely in the stage of preliminary discussion.

"It is perfectly true and only natural that the shipowners and shipping companies should display very considerable interest in a discovery such as that of Flettner, which contains the possibility of a revolution in shipbuilding technique; but at the same time they are no less under the obligation to guard themselves against too hasty conclusions. As things stand at present there can only be a question of carrying the experiments that have been made hitherto a stage further. The fact that the trials so far made with the small experimental ship BUCKAU have yielded good results is no final guarantee that the Flettner sail will stand the test of heavy winds, to which there has not yet been any opportunity of exposing it, or that it will prove to be usefully applicable to ships of larger size. These questions must be settled before one can proceed to the laying down and building of new rotor ships.

"Not less difficult is the situation in regard to the transformation of existing steamers or motorships. A complete elimination of machinery for purposes of propulsion does not seem to be practicable at present, and a combination of machinery and rotor sails depends upon the question of what the effect of the wind would be on the rotating sail of a ship which was already being driven forward by engine power. It is to be expected that some light will be thrown on this matter in the course of the next few months.

"Another decisive point is, naturally, the financial question. The conversion of freight steamers into rotor sailers demands in most cases extensive constructional alterations in the holds, and these must considerably add to the expense. In view of the by no means rosy financial position in which the German shipping companies find themselves, any extensive programme of new construction is absolutely out of the question. It is hoped, however, that in due course of time tramp shipping, which is not tied down to fixed sailing plans, will be able to make use of the rotor sail."

Through parties closely connected with German shipping interests we have been favored with an expression of views that contribute considerably to the understanding of the present state of development of the rotorship. From these sources we learn that while the trials of the BUCKAU have demonstrated that the Flettner system of wind propulsion is practicable, these trials have also shown that numerous modifications still have to be made before the maximum efficiency can be secured. For instance, although the Flettner towers on the BUCKAU are much shorter than those which must be fitted in larger ships, they show a considerable wind loss due to the tendency of the wind to ascend in a spiral course up and around the towers. To overcome this tendency it is planned to install at certain spaced distances up the towers large flat rings similar to the disc at present fitted at the upper extremity of the tower.

It is stated that the electric motors at present used for driving the towers on the BUCKAU are not sufficiently powerful to permit the towers to be rotated at their most efficient speed. The present maximum peripheral speed is about 100 ft. per sec. The pressure effect of the wind on a rotating cylinder reaches a maximum when the peripheral velocity of the cylinder is between 3 and 3.5 times the velocity of the wind. It is proposed to increase the peripheral speed to 150 ft. per sec. on the towers of the BUCKAU, and for this purpose the electric motors will shortly be changed.

It has been pointed out to us that in the event of the vessel encountering a blizzard or operating in low temperature climates where ice or snow is likely to be deposited on the tower surfaces, the efficiency of the towers will be seriously impaired, and the accumulation of snow or ice at the base may even result in a complete stoppage of the towers. Uneven accumulation of ice or snow on the towers would upset the balance of the rotating towers, which are not of very heavy scantlings and unable to resist heavy stresses. The steel plating of the BUCKAU towers is about 1/16 in. thick and is therefore highly vulnerable to damage from cargo or cargo handling. Any damage of this sort would have a serious effect upon the operation, as owing to the high peripheral speed the towers must be perfectly regular in order to avoid the setting up of vibrations.

No cargo handling gear appears to have been fitted on the BUCKAU, and it seems likely that the installation of any derrick masts, or Samson posts on larger ships, would interfere considerably with the flow of wind to the towers. It is likely also that there would be obstacles in the way of handling derrick booms without damage to the towers.

There appears to be no difficulty in the stability question. In the BUCKAU the metacentric height has actually been increased about 10 per cent by the conversion from sailing ship to rotor ship.

There are however many problems presented by the fact that a rotor ship must necessarily have some form of auxiliary power, because in the event that any derangement of the tower operating gear should occur it is not possible to fit a jury rig as in the case of a vessel carrying canvas.

Experience in this and other countries has shown that except when the owner of a vessel is his own master the auxiliary motor is frequently abused. It has never been the practice to carry engineers on auxiliary ships, and there always seems a tendency for the auxiliary engine to be used continuously except when the vessel has the wind astern. Even under the best conditions the master of a rotor vessel would be faced with conditions calling for careful judgment with a view to conserving his fuel, so that any calm periods encountered towards the end of the voyage would not find the vessel unable to use its engine because of lack of fuel.

Indeed, since the rotor ship must, for the reason above explained, have an auxiliary engine unless she is to operate entirely in crowded waters where help could be counted upon, it may be stated that the comparatively large weight of fuel likely to be necessary on extended voyages might place the rotor ship at an operating disadvantage when compared with the standard sailing vessel, because of the cargo deadweight thereby sacrificed.

Time alone will show where the rotor ship will fit into the economics of shipping.

There appears to be substantial ground for the belief that John Brown & Co. of Glasgow are at work on Diesel engines of a larger size than have yet been attempted in commercial work. This machinery is believed to be for installation in the big ship they now have under construction. This company very rarely discloses information about the work it has in hand. In this case as much secrecy seems to surround their new motorship as if she were a new type of war vessel for the British government.

A recent list of motorships equipped with Diesel engines of the Burmeister & Wain type shows a total of 168 vessels with just over half-a-million horsepower and carrying 1,330,000 tons d.w. Only vessels put into service before January 1, 1925, have been included.

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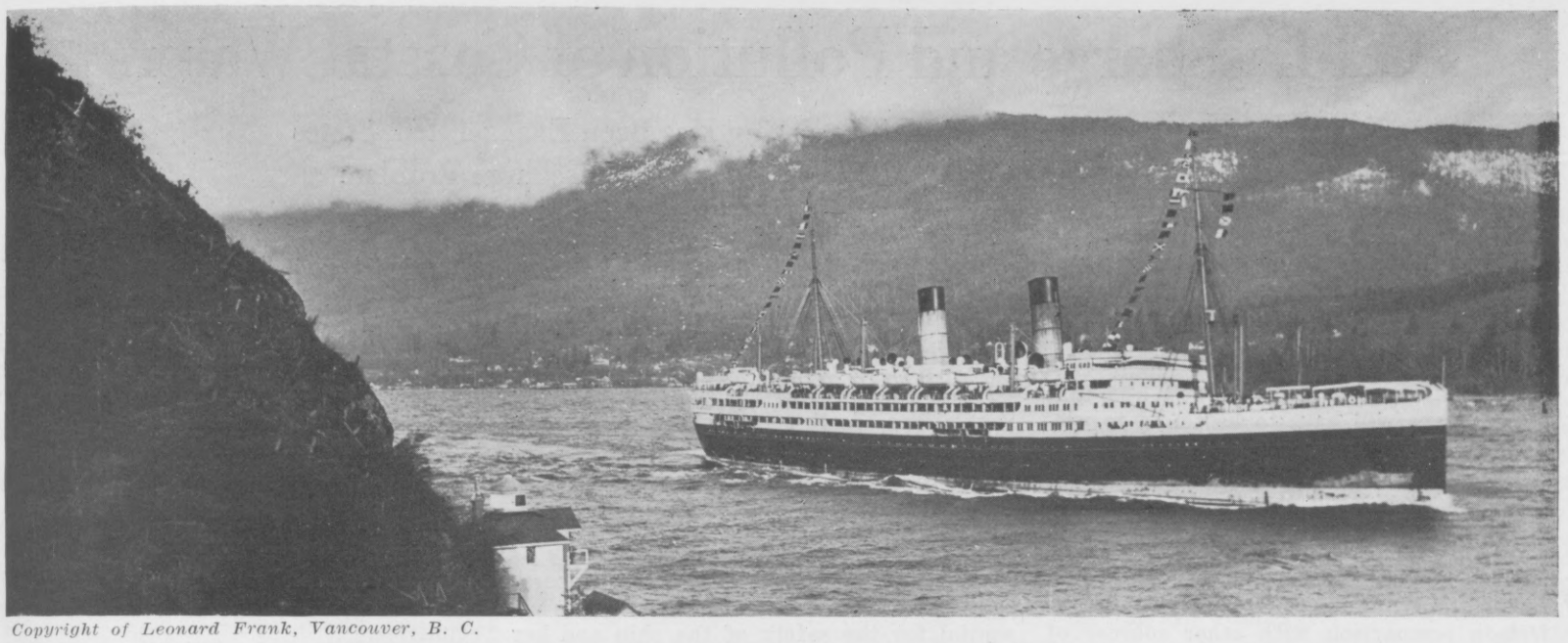

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Transpacific passenger liner AORANGI entering the First Narrows at Vancouver, B. C.

Motorliner Aorangi Makes a Good Start

Successfully Completes a Maiden Voyage of 9000 Miles Before Going on Her Regular Canadian-Australasian Run

WELCOMED by shrieking whistles and vessels dressed with flags, the splendid new motorship AORANGI, the first large passenger liner to have Diesel power, swept into the harbor of Vancouver, B. C., on the morning of January 30th, completing a voyage of over 9000 miles from Southampton via Panama Canal and ready to go on the Canadian-Australasian Royal Mail run out of Vancouver.

This 23,000 tons displacement passenger liner of the Union Steamship Co. of New Zealand, designed for a speed of 18 knots and driven by four 6-cylinder 2-cycle Sulzer-Diesel engines developing a total of 13,000 s.hp. at 127 r.p.m., is 600 ft. long, 72 ft. breadth and 46 ft. 6 in. depth.

Her voyage out was a triumphant one, with airplanes welcoming her to the Pacific at Panama and with enthusiastic greetings at the various ports touched on the trip. Capt. Robert Crawford was in command, with Chief Engineer Henry Lockhart in charge of the engine room.

Her officers state that the ship performed magnificently in spite of heavy weather at the start of the voyage. Her maneuvering qualities proved exceptionally good, and she handled so well in the Panama Canal that she would probably have made a record there if it had not been for delay caused by other vessels locking through.

This being the first voyage of the AORANGI, it was hardly to be expected that the main engines would be driven hard till they had been worked in, and her schedule did not require a speed of over 17 knots. However, she easily picked up 15 hours that she was late in leaving Southampton and that in spite of several days of stormy head winds and heavy seas. During the voyage her engines were ordinarily kept turning at 115 r.p.m., which was 12 revs. below the normal speed for which they were designed. Finally, to show some of the possibilities of speed that lurk in the massive but clean-lined hull and tireless machinery she was opened up to 122½ r.p.m. on the last leg of 85 nautical miles between Victoria, B. C., and Vancouver, doing this distance in 4 hrs. and 24 min. That was 19.4 knots over the ground, but as she had a bit of tide with her the speed through the water was actually about 18 knots.

The fuel consumption for all purposes on the voyage of just over 9000 nautical miles was 1270 tons, less than half of her fuel capacity of 3000 tons, which would thus give her a steaming distance of over 20,000 miles without re-fuelling. This fuel oil, besides being used for the main and auxiliary Diesel engines, was used for steam heating, raising steam for the winches, firing the galley ranges and for running the steam laundry, said to be the largest one afloat.

On one of the best day's runs of 428 nautical miles with a smooth sea and gentle breeze, the ship making 17½ knots, the fuel consumption for all purposes was 54 tons, of which 44 tons were used by the main engines.

Leaving Southampton January 2nd the AORANGI had 400 passengers, 250 in the first class and the remainder in second class. The only cargo was 1630 tons of pig iron that had been loaded in addition to water ballast to put her down to her load line for the trial trip, her mean draft on leaving being 26 ft. 4 in.

A delay of 15 hours in getting away from the Solent was the result of not being able to put the pilot overboard in his small boat in the prevailing bad weather. For the next three days of the run to Kingston, Jamaica, strong head winds and heavy seas prevailed, but good weather followed. This was the longest non-stop run, 4323 nautical miles, the time being 266 hours from light to light. The next stage of the trip, from Kingston to Colon, was 553 miles, with strong breezes and moderate swell. This run was made in 35 hours and 40 minutes. Between 9 hours and 9½ hours were spent passing through the Panama Canal, there being a delay of 1½ hours at Miraflores lock and half an hour's delay at Pedro Miguel lock on account of other shipping.

From Balboa, at the Pacific end of the canal to San Pedro, a distance of 2920 miles, was made in 175 hours and 37 minutes, with moderate to light winds. The next stage from San Pedro to San Francisco, 369 miles, was made in 27 hours and 17 minutes with smooth sea and heavy fog, in which the ship had to slow down, Captain Crawford being 20 hours on the bridge at one stretch. Then came a 753 mile stretch from San Francisco to Victoria in fog and rain, made in 45 hours and 24 minutes, and a run of 85 miles from Victoria to Vancouver in 4 hours and 24 min. Times are from port to port, and do not take into account maneuvering in and out of harbor. About three days was the total spent in the different ports en route, less than a day being spent at Kingston, about a day each at San Pedro and at San Francisco, and overnight at Victoria.

AORANGI's equipment, from the main engines to the electric boat winches on the davits and to the gyro-pilot for automatic steering, indicate improvements and advances in many lines.

Capt. Crawford has a pardonable pride in his fine command, and looks well after the vicissitudes of the voyage, which included being captured and photographed in the center of a bevy of San Francisco girls, after being 20 hours on watch coming through the fog banks to the Golden Gate. He speaks highly of the maneuvering qualities of the AORANGI and of her general seaworthiness. Mr. Henry Lockhart, the Chief

(*Continued at bottom of next page*)

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Oil Discharge and Pollution of Coastal Waters

Factor of Shipping in Oil Pollution Has Been Exaggerated, States British Committee in Reporting on Oil Discharge Problem

CONSIDERABLE concern has been caused in oil engineering circles by the opinion expressed by the Oil Discharge Committee of the British Chamber of Shipping and the Liverpool Ship Owners' Association in its recently issued report that oil separators are unsatisfactory and that other means will have to be found to prevent the pollution of the sea coasts and coastal waters by fuel oil discharged from ships.

The committee expresses the opinion that while oil discharged from merchant vessels must be taken into account as one of the factors contributing to oil pollution, its effect in comparison with other sources of pollution has been much exaggerated. From analysis of the replies received from shipowners to a questionnaire, it has arrived at the conclusion that the more serious source of oil discharge from merchant vessels arises from oil-tank vessels when being cleaned out preparatory to drydocking for hull repairs, or in preparation for fresh cargo, and from vessels carrying fuel oil and water ballast alternatively in their double-bottom or other ballast tanks.

Effect of Wind and Tide

No evidence was submitted to show whether oil may safely be discharged at a considerable distance from the shore, nor, of course, as to what that distance should be. If it is found by experience that this matter can be further considered internationally:

"As so much depends on the strength and direction of such variable influences as the ocean currents, tidal streams, and prevailing winds in the locality in which the discharge takes place, no general rule can be laid down as to the distance off shore at which oil may be safely discharged, but it is probable in any area up to a distance of 150 miles from land, where the resultant of current tidal streams and drift due to wind happens to produce a general onshore drift, discharged oil will eventually reach coastal waters and other indentations of the land."

The committee further suggests that any regulations dealing with the high seas should incorporate a provision for compulsory records in ships' logs of pumping operations of oily water within the limit mentioned. Care should be taken to exclude normal pumping of bilges, the quantity of oil in which is negligible, and which operation in many cases is automatic and is essential for the safety of the ship and her cargo.

Requirements for Oil Separators

The report gives the results of the committee's detailed examination of oil separators as follows:

(1) To be of economic value a separator on board ship should not only free the oily water of the oil, but should free the oil so separated from water and also from dirt.

(2) The cost must be reasonable, having regard to the initial cost of the separator (including fitting), the cost of repairs, maintenance and operation, and the speed at which it does effective work.

(3) The space occupied, location, and weight of separator must be arranged to suit individual requirements on board ship.

(4) Oil separators in their present state of development are not capable of dealing with certain mixtures of oil and water, unless used in conjunction with settling tanks with heating coils or other apparatus.

Recommendations for International Action

With regard to the methods of dealing with pollution in the case of merchant ships, the committee admits that it will be difficult to identify an offending vessel by circumstantial evidence. It suggests that an essential element in the solution of the problem is that all countries who have not already done so should put themselves as regards their territorial waters in the same position as the United States and Great Britain by passage of legislation.

The committee's recommendations are as follows:

(1) That the compulsory fitting of separators is undesirable.

(2) That identical regulations should be agreed to and enforced in all maritime countries.

(3) That such regulations should apply only to crude petroleum, fuel, and Diesel oil.

(4) That subject to (3) above, such regulations should prohibit the discharge of oily ballast water and tank washings by all vessels other than naval craft within 150 miles from the nearest coast.

(5) That such regulations should be enforced by a compulsory record in the ship's log of pumping oily ballast water and tank washings.

(6) That naval authorities in all countries should impose on naval craft corresponding regulations.

(7) That port authorities throughout the world should be responsible for providing, either by themselves or through third parties, separating scows or other receptacles in adequate numbers whenever the port is used by oil-driven or oil-carrying vessels, where such oil falls within (3) above.

The report is signed for the committee by the chairman, Archibald Maclean, of the Anglo-American Oil Co. (the British subsidiary of the Standard Oil Co.) The other large oil companies and the leading shipowners were represented on the committee.

Motorliner "Aorangi" Makes a Good Start

(*Continued from previous page*)

Engineer, was highly pleased with the way the engines worked, the lack of trouble and the good shape of the cylinders when pistons were drawn for inspection at Vancouver.

The AORANGI will operate on the run from Vancouver, B. C., to Sydney, N. S. W., calling at Honolulu, Fiji Islands and Auckland, N. Z., on a schedule that calls for a sea speed of 18 knots. She compares favorably in point of size, speed and passenger accommodations with the finest liners operating on the Pacific Ocean and is the largest of the fine fleet of steamers and motorships operated by the Union S. S. Co., carrying freight and passengers on the Canadian-Australasian run.

The C. P. R. liner EMPRESS OF CANADA, running from Vancouver to the Orient and the present holder of the speed record for the Pacific among merchant ships, is somewhat larger than AORANGI, being 653 ft. o.a. by 77 ft. 9 in. beam. She is equipped with geared turbines driving twin screws developing 23,000 s.hp., and was designed for a normal sea speed of 18 knots. The EMPRESS OF AUSTRALIA, another C. P. R. liner on this Oriental run, is 615 ft. o.a. and 75 ft. breadth, with a load displacement of 24,950 tons. She is fitted with turbines working through hydraulic reduction gears of the Föttinger type and developing 14,500 s.hp. Her speed is about 17 knots. She was built by Vulcan Werke of Stettin, and turned over under the terms of the Peace Treaty.

The high standards of luxurious accommodation and seaworthiness set by these liners and by those on the Australian run have been followed out and improved on in some particulars in the design and fitting out of AORANGI, which is in all respects a worthy pioneer of the large motor liner for long ocean passages.

Summary of Data on Maiden Voyage

SOUTHAMPTON TO VANCOUVER

Distance by observation	9047 nautical miles
Average speed maintained	17 knots
Best day's average speed	17.5 knots
Average revs. on best day	118.8 r.p.m.
Average revs. on voyage	115.5 r.p.m.
Average power on best day	13,200 i.hp. = 10,500 s.hp.
Average m.e.p.	88½ lb. per sq. in.
Total fuel consumed at sea	1242 tons
Total fuel consumed in port	24 tons
Total fuel for best day's run	54 tons
Main engine fuel for best day	44 tons
Distance of longest non-stop run	4323 miles
Number of hours light to light	266 hours
Gravity of fuel oil used	26.5° Beaumé
Number of men in engine room	23 engineers, including 4 electricians and 1 refrigeration engineer, plus 22 oilers, donkeymen and storekeepers.
Make of oil purifier	2 Sharples 300 gal. p. m. and 1 De Laval 150 gal.

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Giant Ore Carriers of 21,000 Tons Deadweight

Two Motorships of This Capacity Have Been Launched and Will Carry Iron Ore from Chile to New York

ABOUT the end of March the two biggest freighters in the world will be commissioned under the house flag of Axel Broström, a Swedish Corporation which has been responsible for a lot of progressive motorship ideas. These two freighters will be propelled by Diesel engines of 4500 hp. aggregate and will be frequent visitors to New York, having been chartered to carry iron ore from Chile for the Bethlehem Steel Company's furnaces. They have been christened the SVEALAND and AMERIKALAND.

Iron ore, being very heavy, stows in a small space. The whole 20,000 tons of ore which each of these vessels will carry will be loaded into the relatively small hold space provided above a double bottom of abnormal depth. This arrangement has been adopted with a view to obtaining a small metacentric height to secure good and quiet behavior in a heavy sea. The ore holds are 30 ft. wide and 30 ft. high with a total length of 365 ft. Their bottom is 14 ft. above the keel plating.

Several pictures taken during the progress of construction at the Deutsche Werft, Hamburg, are reproduced below to show the construction of the hull and give an idea of the special design. It will be seen that very large ballast tanks are provided on both sides, and the exceptional height of the double bottom is also very evident. The first picture was taken Feb. 24, 1924, and the last picture on Jan. 24 of this year, at the time of the launching of the SVEALAND, while the AMERIKALAND was still on the ways.

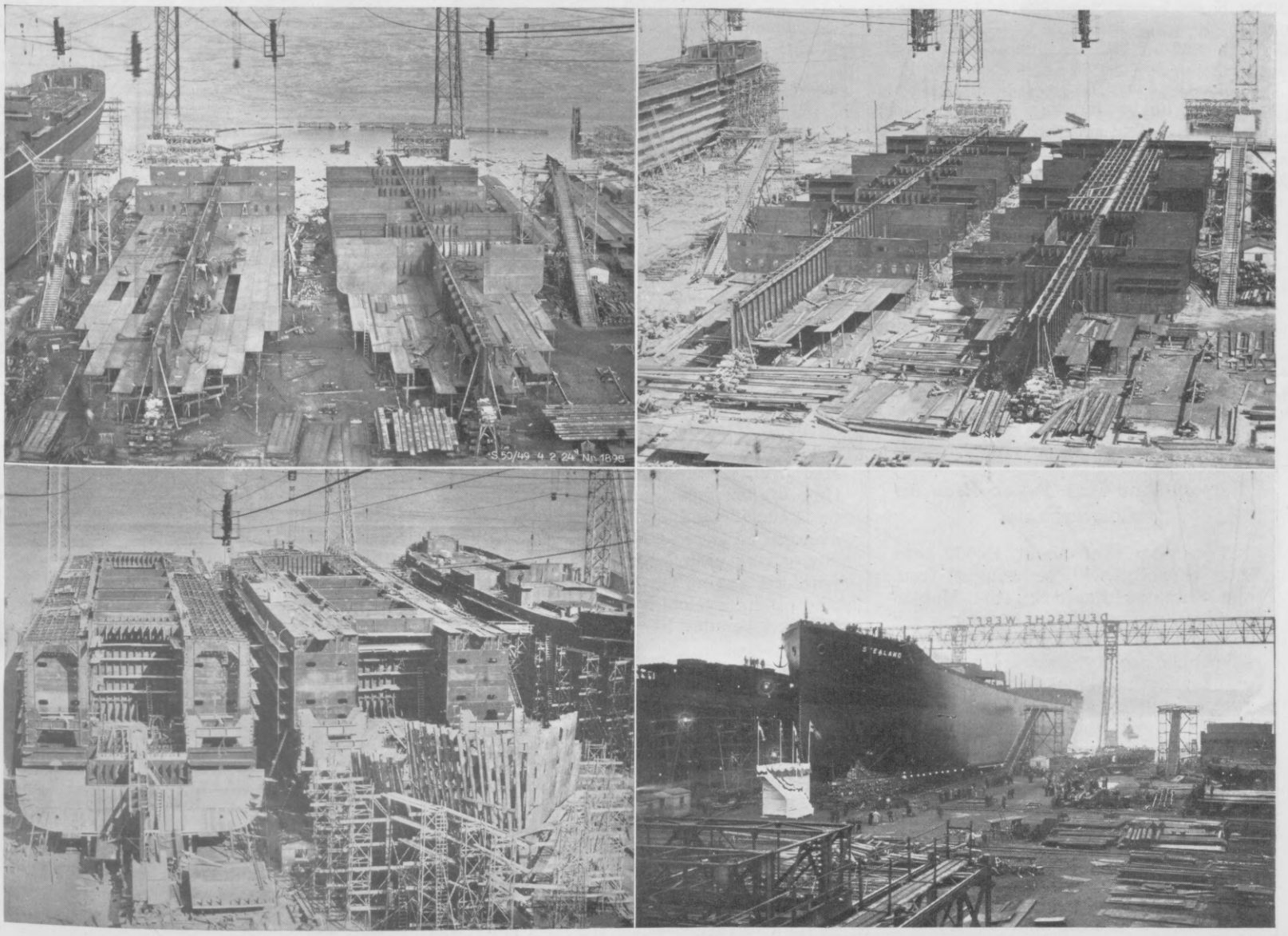
The principal dimensions of these two vessels, which are sister ships, follow:

Length over all	572 ft. 0 in.
Extreme breadth	72 ft. 0 in.
Molded depth	44 ft. 0 in.
Loaded draft (summer freeboard)	32 ft. 4 in.
Block coefficient	0.81
Displacement, loaded	29,500 tons
Deadweight capacity	21,000 tons
Cargo capacity	20,000 tons
Launching weight	7,500 tons
Total power (twin engines)	4500 hp.
Engine speed	115 r.p.m.
Ship's speed (loaded)	11½ knots

These vessels, which will be propelled by twin screws, will be driven by two single-acting engines of the Burmeister & Wain type built by the A. E. G. The auxiliary machinery consists of two Diesel-driven generator sets of 100 kw. each, which will also provide the necessary air for maneuvering the main machinery. All the auxiliaries in the engine-room and on deck will be electrically operated.

SVEALAND is expected to make her trials early in April, and AMERIKALAND will be finished up about one month later. These unusual vessels embody as far as practicable in the design of ocean-going ships the hold arrangements of the Great Lakes ore-carriers which are unmatched for quick loading and discharge.

At the former German Navy Yard in Kiel, now operated by the Deutsche Werke A. G., there are six motorships under construction. The biggest is a cargo vessel of 11,300 tons and of 3600 hp. for the Hansa Line. There are also in hand a 6400 tons freighter for W. Wilhelmsen, a 6000 tons vessel for Fearnley & Eger of Oslo, and three vessels of 1700 tons each for the Neptune Company at Bremen.



Progress of construction on ore-carriers of 21,000 tons d. w., showing ore-holds, side tanks and double-bottom

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Notes and News of the Month from Everywhere

Record of New Construction, Launchings, Trials, Ships' Performances and Other Events at Home and Abroad

For the new tug DONALD M. MCNEALE of the Smoot Sand & Gravel Corp., Washington, D. C., a Lombard engine has been ordered.

At Brewer's Drydock the steel tug CLIPPER is being converted from steam to motor power for the Walnero Towing Co. of New York. Her engine will be a 6-cylinder 150 b.hp. C.O. type.

Four boats will be operated by the South Shore Port Company from the lower end of San Francisco Bay to the city and to Oakland. These boats will be 110 ft. long and will be propelled by twin-screw Atlas Imperial engines of 90 hp. each.

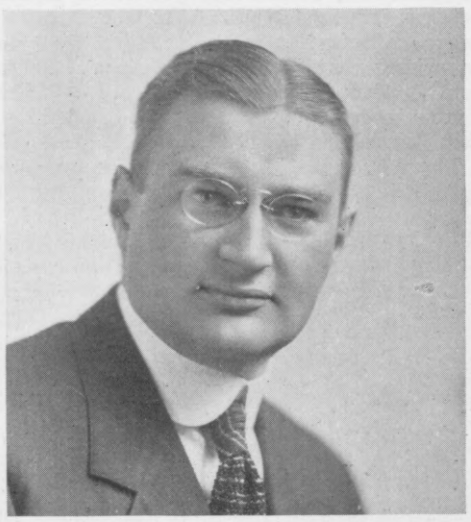
All of the 325 Lake-built vessels of the Shipping Board have been offered for sale at substantial price reductions. Any of them can be purchased at a price of $50,000 each, less the sum estimated by the Board to be necessary for reconditioning, so that a ship requiring repairs estimated at $15,000 would be available for purchase at $35,000. The boats range from 2875 d.w. tons to 4350 d.w. tons.

When the New York State Barge Canal opens for navigation again in the spring the Connelly Transportation Co. of Buffalo will have three motor tugs ready for service. One of these is the W. F. MATTICK, formerly owned by the Globe Shipbuilding Co. of Buffalo which originally constructed her for its own account. She was described in the December issue of MOTORSHIP. The two other boats will be of about the same size. The hull of one of them has been completed and the second hull is now in frame. These new boats, like the W. F. MATTICK, will be powered with Fairbanks, Morse engines of 100 s.hp. turning at 250 r.p.m. The boats are 65 ft. in length.

Matson Line Has Taken Bids on Conversions

Two vessels of about 11,000 tons d.w. capacity will be bought from the Shipping Board by the Matson Navigation Co. of San Francisco for conversion to Diesel power if conversion bids are favorable. These two ships are the COURAGEOUS and TRIUMPH, built by the Bethlehem Shipbuilding Corp., and of about 7600 tons gross. The former was completed in 1918 and the latter in 1919. Their registered dimensions are 440 ft. by 56 ft. by 35 ft. 2 in., and the present power is derived from geared turbine installations. It is planned to replace this equipment by Diesel engines of 2800 hp. to be built by the Wm. Cramp & Sons Ship & Engine Building Co.

Gordon S. Rentschler, who was president of the Hooven, Owens, Rentschler Co., was honored in January by being chosen as a vice-president of the National City Bank of New York, with the duty of assistant to Charles Mitchell, president of the bank. Mr. Rentschler had been on the Board of Directors of the National City Bank since October, 1923, and his graduation from industry into such an important position with the biggest bank in the country has seldom been paralleled. Mr. Rentschler, who is only 40 years of age, is the youngest of the bank's directorate. At the last annual meeting of the Board of the Hooven, Owens, Rentschler Co., Mr. George A. Rentschler was elected president of the company and Gordon S. Rentschler appointed chairman. Fred A. Rentschler,



Gordon S. Rentschler

formerly president of the Wright Aeronautical Corp., was made a director and vice-president of the company.

CUBORE, the big ore-carrying vessel driven by Bethlehem 2-cycle engines, has covered a total of 73,800 miles in the last year. Her average mileage during the elapsed time of 369 days has been 200 miles per day, including loading and delays. The average mileage at sea was 243 miles per day, giving an average speed of 10.3 knots with the engine turning at an average of 99.4 r.p.m. developing 2400 s.hp. The average fuel per day has been 12.05 tons.

HAWAIIAN STANDARD, the third motor tanker of the Standard Oil Co. of Cal., has been launched at the Bethlehem yard in San Francisco. She is a single-screw vessel with a deadweight capacity of about 9000 bbl. and is intended for the Hawaiian service. She is very similar to the STANDARD SERVICE and ALASKA SERVICE owned by the same company, the chief difference being that the two Pacific Diesel engines are being installed amidships in the new vessel whereas in the two former boats the machinery was placed aft.

At the first February meeting of the Propeller Club of New York an interesting lecture on the Flettner rotor ship was delivered by F. O. Willhoft, treasurer of Th. Goldschmidt Corp., who are the agents in this country for the Flettner invention.

At the Naval Academy at Annapolis the official tests of the 100 hp. Knudsen engine have been completed. They included an endurance trial of 50 hours at full load.

Engineers having refrigerating machinery in their care will be interested in a short article on the lubrication of such machinery contributed by the technical department of the Vacuum Oil Co. to its house organ, the *Compass*.

Capt. Tellessen, formerly connected with one of the Barge Canal companies that operate motor fleets, will have his own tow boat in service on the Canal next season. The hull is being built at Norfolk, Va., and will be equipped with a 6-cylinder 150 hp. C-O. type of Fairbanks, Morse engine.

In his speech before the Naval and Marine Memorial Association at its dinner in New York on Feb. 18, Mr. O'Connor, Chairman of the U. S. Shipping Board, made the following remarks: "Wherever they can get it printed they constantly speak of our national loss in keeping up our merchant marine. Loss is not the proper word. The proper word is cost. Does anyone refer to government money spent on roads and highways as loss? No, they call it cost. . . Congress appropriates money to maintain the Department of Agriculture. Does anybody dare call that appropriation a loss? The Post Office Dept. spends money upon Parcels Post and Rural Free Delivery. Does anybody dare to call that expenditure a loss? Transportation companies attacked the Parcels Post at one time, but they quit and quit forever."

Ford Motor Co. Will Convert Large Steamer

A ship of about 11,700 tons d.w. capacity has been purchased from the Shipping Board by the Ford Motor Co. of Detroit for conversion to Diesel power and will be used by that firm in ocean transport. This vessel, EAST INDIAN, was built in Japan during the war for the United States Government and is of 8229 tons gross. Her registered dimensions are 445 ft. by 58 ft. by 29. ft. She is a twin-screw boat with triple expansion engines. New machinery equipment has not yet been fully decided on, but the reconditioning of the vessel has been intrusted to the Sun Shipbuilding Co., and she is now at their yard at Chester, Pa.

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Driven by an oil engine, this converted sub-chaser RELIABLE is now used for fishing party excursions out of New York

One of the latest additions to the fleet of deep sea fishing party boats operating out of New York is the RELIABLE, formerly a 110-ft. submarine chaser and now converted to oil engine power. Her original machinery has been replaced by a 6-cylinder 135 hp. Standard airless injection engine which gives her a speed of about 13 miles an hour at a fuel cost of just over 50 cents an hour with the present price of fuel. Her owner is Capt. Edward Baletti, who runs her from Hoboken, N. J., and the Battery Park landing, New York, to the fishing banks off the Jersey coast. A picture of her is given on this page.

At the Harlan plant of the Bethlehem Shipbuilding Co. the former steam tanker ANAHUAC has been converted to an oil-engined vessel and renamed BRILLIANT. She measures 180 ft. by 31 ft. by 17 ft. and has a capacity of about 10,000 bbls. Her machinery consists of two 225 b.hp. Ingersoll-Rand oil-engines driving d.c. generators which furnish power to a 375 hp. electric motor turning at 120 r.p.m. The electrical machinery was built by the General Electric Company.

Some months ago mention was made in these columns of the smallest Diesel electric drive, referring to an installation with a Bull Dog oil engine made by a shipping company in France. The experimental installation has proved satisfactory, and the company has decided to equip its entire fleet of small barges with a similar drive. An order has been placed with the Bates & Edmonds Motor Co. for nine Bull Dog engines of 18-20 hp. and four engines of the same make rated at 12-14 hp. Generators of French manufacture will be used.

Consideration is being given by the Administrative Council of the Atlantic Coast Shipbuilders' Association to the advisability of establishing a standard form of contract for the shipbuilding industry. It was pointed out that considerable waste and litigation is now occasioned by the indefiniteness of some contracts. It is contemplated that contracts should be standardized for new construction and for repair work. The Association's office has on file copies of contract forms used by the members of the Association and those adopted by other business and industrial organizations.

The Union Oil Co. of Cal. is increasing its storage capacity for Diesol at the Panama Canal from 37,000 bbl. to 74,000 bbl. and plans to carry in stock a regular Diesol of about 26 or 27 deg. Beaumé and a lighter grade for small boats that do not get good results from the standard grade. The oil company's price at the beginning of February was $2.15 per bbl. for pipeline delivery at the oil dock.

For the new automobile ferry to be operated by the Crosby Direct Line Ferries, Inc., from Alki Point to Manchester on the Olympic Peninsula one of the 600 s.hp. oil engines built by the H. W. Sumner Co. during the war period for the Chilean Government has been selected as the motive power. The ferry was designed by A. H. Coolidge of Seattle, who will also superintend the construction of the ferry in the yards of the Crosby Marine Corp. at Salmon Bay. The new boat will be able to carry 65 automobiles and will cost approximately $250,000. It is stated that the engine was selected on the basis of the average price of the single-acting engines in the awards recently made by the U. S. Shipping Board for the conversion of its eighteen vessels selected for this purpose. This means that the Sumner engine would have fetched $84.70 per s.hp.

Oil-engined tanker BRILLIANT, of the Atlantic Refining Co., converted from steam power

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AORANGI was struck by a heavy wave the day after leaving Vancouver on her first voyage to New Zealand, with injury to two passengers and damage to the rails.

For the Union Steamship Co. of New Zealand, owners of the AORANGI, a twin screw motorship 240 ft. long is to be built on the Tyne. The same owners are believed to have another motor vessel under consideration.

It is a better gamble that the AORANGI will have to measure up against oil-burning steamers in the next succeeding years—states the *Shipping Register* of San Francisco—than it is that she will be forced to demonstrate her economy as compared with a coal burning steamer.

BELDIS, a vessel of 3450 tons d.w. built last year for the A. S. Christen Smith Rederi of Christiania and powered with a 4-cylinder Sulzer engine of 1350 hp., recently loaded 17 locomotives of 70 tons each consigned by an English firm to one of the Indian railroads. The vessel was specially designed for such shipments.

A 9000 tons d.w. single-screw vessel for the Hopemount Co., Ltd., has been launched by Swan, Hunter & Wigham Richardson. The vessel measures 400 ft. b.p., with an extreme breadth of 55 ft. 3 in. and a depth moulded to second deck of 29 ft. 6 in. She will be driven by a 6-cylinder Neptune 2-cycle engine.

About the end of January the Bank Line accepted from Harland & Wolff the twin screw motor vessel CLYDEBANK, 434 ft. long, 53 ft. 9 in. in breadth and 37 ft. in depth with a gross tonnage of about 5200 tons. Her propelling machinery consists of two sets of Harland-B. & W. 6-cylinder engines. The engineroom auxiliaries are electrically driven as also is the steering gear, but the cargo winches and windlass are steam driven. In addition to the double bottom, which can be used for oil fuel, No. 3 hold has also been arranged for transportation of oil fuel or oil cargo.

Another motorship, the SILVERRAY, will be put into service in June by the Kerr-Roosevelt Line, making the sixth of the fleet of eight 8400 tons motor-vessels employed by them in round-the-world trade.

ALYNBANK has been launched by Harland & Wolff for the Bank Line. She is a shelter deck vessel of about 5200 tons gross, measuring 434 ft. x 53 ft. 9 in. x 37 ft. Her machinery will be two sets of 6-cylinder Harland & Wolff engines of the B. & W. type. She is the first of a number of motorships which will be launched for the Bank Line this year.

CLYDEBANK, recently completed by Harland & Wolff for the Bank Line, was the ninth vessel of the big order which those builders received from Andrew Weir & Co., who are the managers for the Bank Line. ALYNBANK, which has been launched, is the tenth vessel under that contract and the ELMBANK, launched middle of February, is the eleventh. It is understood there are nine more to be completed.

In the January issue of *The Shipyard*, the works magazine of Swan, Hunter & Wigham Richardson, Ltd., appears a tabulation of the firm's output in 1924. It is there shown that the combined tonnage launched by that firm and its associates during 1924 was greater than that of any other firm or group of builders in the world. The list includes seven motorships and two motor yachts.

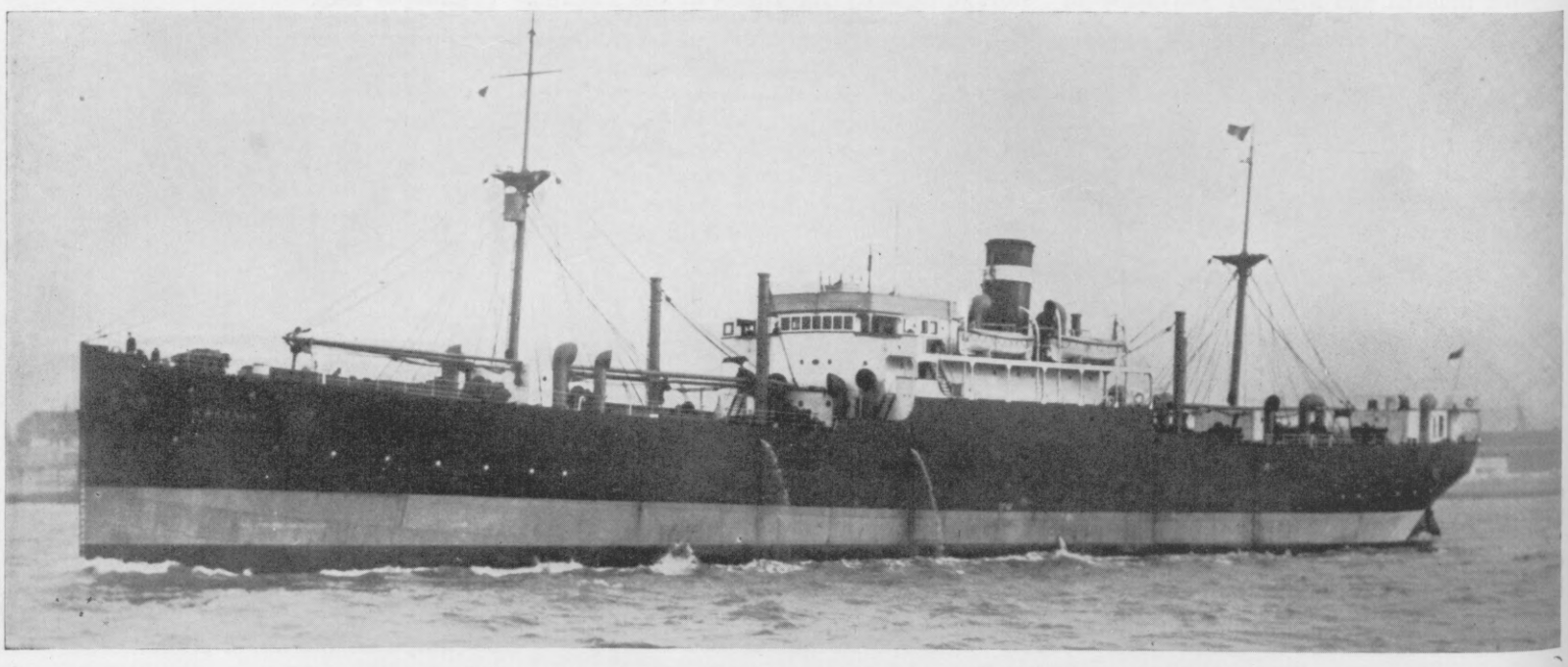
Fairplay of London has published the following dimensions of the big motor liners now building at the Harland & Wolff yard in Belfast. The Union-Castle Liner is to be 630 ft. by 73 ft. by 46 ft. and will measure about 20,000 tons gross. She will be known as the CARNARVON CASTLE. The R.M.S.P. boats are each to be 655 ft. 8 in. by 78 ft. 6 in. and of about 22,000 tons gross. All three ships will have similar machinery, the main equipment in each case consisting of two sets of 8-cylinder double-acting 4-cycle Harland & Wolff engines of the B. & W. type developing together about 20,000 i.hp.

A Fiat engined vessel of 8000 d.w. tons has been launched at Monfalcone for the Cosulich Company. This vessel has received the name MAULY. Her propelling machinery will consist of two Fiat 4-cylinder 2-cycle engines, each of 1250 hp. turning at 115 r.p.m.

It is reported that if the Italian government authorizes the proposed subsidy to the Florio firm of Sicily, this company will build ten or twelve motorships for passenger and cargo service on a fast schedule between Naples and Palmermo.

At the Cantiere San Rocco at Trieste the passenger and freight motorship ESQUILINO was recently launched for the Lloyd Triestino. She will measure about 10,000 gross tons, will be 450 ft. in length and 57 ft. in breadth. For propelling machinery two Triestino engines of 5000 i.hp. aggregate will be installed, giving the ship a speed of about 13 knots. These engines are of the Burmeister & Wain type with six cylinders of 29.13 in. diameter and 45.28 in. stroke turning at 115 r.p.m. These engines will be fitted with supercharging. A sister ship, VIMINALE, is building for the same owners. Both vessels will run between the Adriatic and the Far East.

It has not hitherto been quite clear what the size of the new Cosulich motor liners actually would be. Statements have been published that these vessels would measure 25,000 tons gross. If that had been true they would have ranked as the largest combination passenger and freight boats in the world with the exception of the S.S. ADRIATIC and S.S. BALTIC. As a matter of fact, the plans for the Cosulich motor liners are not yet completed. As at present arranged they will be of 25,000 tons displacement, 610 ft. long and driven by two double-acting engines of 10,000 i.hp. each. The propelling machinery will be of the Burmeister & Wain type built by the Stabilimento Tecnico Triestino.



MOVERIA, recently completed for the Donaldson Line, is a single-screw ship of 2700 hp.

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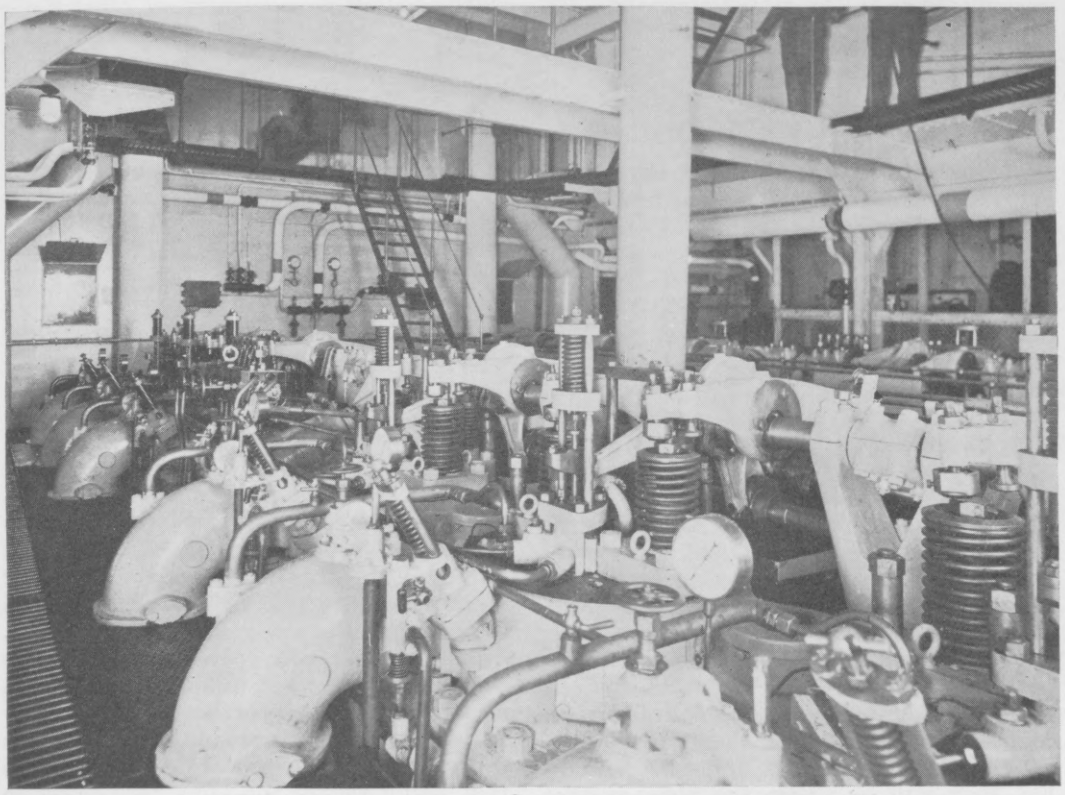

Werkspoor engines now have downward exhaust elbows—VARANGER'S engine-room

One of the largest oil tankships afloat, with a deadweight capacity of 12,750 tons and driven by two Werkspoor Diesel engines of 1800 hp. each, has been delivered to Westfal, Larsen & Co., Bergen. She was built in Holland on the Millar system of framing and has a length of 470 ft., breadth 60 ft. and depth 38 ft. 6 in. She has been named VARANGER and is a twin-screw vessel with a speed of 11 knots.

A motor-tanker of 6500 tons d.w. for the Stoomvaart Mij. de Maas has been launched at the Fijenoord yard at Rotterdam. She will be known as the KATENDRECHT and is 370 ft. long, with 53 ft. breadth and 28 ft. depth. A single screw engine of the M.A.N. type built by Fijenoord and of 1850 hp. will give her a speed of about 11 knots.

NORDANGER, a sister ship of the VARANGER, has been launched in Holland for Westfal Larsen & Co., Bergen. She measures 470 ft. in length, 60 ft. in breadth and has a depth of 38 ft. 6 in. The d.w. capacity is 12,750 tons. Two Werkspoor engines of 1800 h.p each will drive the twin screws.

VARANGER, the latest motor tanker completed to the order of Westfal Larsen & Co. of Bergen has been chartered by a U. S. oil company for 10 years.

Two motorships have been ordered by De Koninklijke Paketvaart Maatschappij of Holland for its Far Eastern service. One vessel has been ordered from the Netherlands Shipbuilding Co. and the other from the De Schelde firm. Both ships will measure 380 ft. by 52 ft. by 27 ft. 6 in. and will be of about 4650 tons gross. The first ship will be named OP TEN NOORT and will have two Werkspoor engines of 2330 hp. each. The second ship will be known as VAN HEUTZ and will have Sulzer engines of the same power.

Another ship for the Norway-Mexico Gulf Line, which is one of the W. Wilhelmsen companies, has been launched by the Deutsche Werke at Kiel. She is the TOPEKA of 6400 tons d.w., with a designed speed of 12½ knots, 365 ft. long, 51 ft. beam and a depth of 34 ft. Two Diesel engines of the shipbuilders' own design will be installed and develop about 2000 s.hp.

CONSUL HORN, one of the new motorships of the Horn Co. of Lubeck, sailed on her maiden voyage on the last day of January bound for the West Indies. She has a Flettner rudder.

Krupps' have launched for the Bugsier, Reederei & Bergungs A-G. of Hamburg the salvage tug WOTAN. She is 142 ft. in length and 26 ft. 3 in. in breadth, and will be propelled by two 4-cycle engines of 1200 hp. each. The contract speed is 15½ knots.

At Krupps yard in Kiel the last of the four motor-vessels for the Horn Company of Lubeck has been launched. She has been christened MARIE HORN and is of 4,000 tons d.w.

For the two Stinnes motorships of 11,000 tons each under construction at the A. G. Weser in Bremen and at the Bremer Vulkan, respectively, the machinery has been ordered from the Hamburg Vulcan firm.

A motor tug-boat of 500 hp. with a hydraulic transformer of the type built by the Hamburg Vulcan company, is under construction at the Bremer Vulkan, Vegesack.

In the two new motorships of the Hamburg South-America Line, named MONTE SARMIENTO and MONTE OLIVIA, a means of comparison will be provided for the value of propeller guide blades. The MONTE SARMIENTO has none, but the MONTE OLIVIA will be fitted with them.

As an example of the changes that are coming over river traffic as the oil engine breaks into that domain, it is interesting to note the order for six 725 d.w. towboats for the Rhenustransportgesellschaft of Basle, Switzerland. These carriers are for traffic on the Rhine. The first of them has been delivered. JUPITER, as she is known, measures 256 ft. between perps., 35 ft. 5 in. beam, and 10 ft. 7 in. depth. Her motive power consists of two M.A.N. 6-cylinder airless-injection engines developing each 200 hp. at 250 r.p.m. and driving twin-screws. A speed of about 10 miles per hour can be attained when the boat is fully loaded. For auxiliary needs there is a 45 hp. 2-cylinder M.A.N. airless engine, driving a d.c. generator to provide current for lighting and for all the engine room pumps and deck winches.

Chartered by General Petroleum Corporation of Los Angeles, this new Dutch motor tanker, VARANGER, can carry 12,750 tons d. w.

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Readers are invited by the Editor to submit articles, photographs or drawings relating to motorships, marine oil-engines or auxiliaries. Contributions used in the magazine are paid for on the 15th of the title month of issue, and other contributions are returned as promptly as possible.

Steamer Conversions Increase Steadily

As time passes it becomes more and more evident that the importance of the motorship in the domain of American shipping is growing at a much faster rate than the figures of new tonnage indicate. First one American shipowner and then another is being added to the list of those who are converting their steamers into motor vessels. The tendency for this change has been more marked among American shipowners than under foreign flags. It began to take root here pretty firmly a couple of years ago and received quite an impetus last Fall when the Shipping Board took action to convert 18 vessels of its fleet. One of the big oil companies announced its decision early this year to install Diesel engines in two of its most modern tankers. Three steamers belonging to the same interests, but operating under the house flags of foreign subsidiaries, are similarly to undergo conversion. Another big oil company is expected to announce its plans for the alteration of one of its steam tankers to motor power during the next month. An addition to the list is also made by a ship, probably the forerunner of others under the same ownership, to be put into operation by a prominent automobile firm for the transportation of its own products to foreign lands. There is likelihood of two conversions to the order of one of the companies operating in the Pacific. If the later order be placed, it will bring to a total of 27 the number of American steamers undergoing conversion at the present time. This volume of reconstruction is practically hidden from the view of the nation at large because it is not covered by the shipbuilding statistics issued by this Government or by the foreign classification societies. The American Bureau reports it in the bi-monthly review of American shipyard work, but the space given by the newspapers to the American Bureau reports is meager. Gradually the importance of this conversion work will be brought home to the statisticians, and the proper place will be given to it in shipbuilding returns.

Biggest Freighters Will Be Motorships

A big advance has been made by the motorship when it can claim for its own not only the biggest tanker in the world, but also the two largest freighters. More than ten years have passed since Diesel engines were installed in a tanker of 15,500 tons d.w., which still holds the record for size among oil-carrying vessels of the world. Two huge freighters of 21,000 tons d.w. capacity, now nearing completion in Europe, will be the biggest freight carriers on the ocean. It is true they are not as big as the largest combination passenger and freight vessels, but they will head the list of straight cargo carriers. They are to be driven by Diesel engines and thus add another record to motorshipping. Report has it that these giant carriers have been chartered by an American firm for a period of twenty years, which means the whole of their useful life as figured by accountants. Inasmuch as they will be running between New York and South America, they will be under the view of that large section of American shipping interests located in the Port of New York and will be a potential source of inspiration to the shipowners of the Atlantic Coast.

Oil Pollution in Coastal Waters

Even though shipping circles would prefer that the subject of oil pollution in the waters adjacent to the coast could be buried forever, there are a sufficient number of interested people to keep the matter alive. There is no reason whatever why the evil should not be frankly faced. The shore resorts undoubtedly suffer from films of oil spreading over the surface of the water, interfering with the pleasure of great numbers of holiday visitors and residents, and constituting a nuisance. Motorships are less serious offenders than steamers, but even so the contribution of motor vessels to this pollution should be terminated so far as practicable and as early as possible. With all the aspects of the problem of eradicating this evil we need not deal. Part of the trouble is caused by factories on the water front. The oil tankers are stated to be serious offenders. Vessels of every description under power are undoubtedly contributors, and under certain conditions of wind and tide there is more oil carried inshore than may come out of the estuaries and river mouths. On another page of this issue is published a summary of the report of a committee which investigated the subject in a foreign maritime country. The conclusions and recommendations are worthy of study even if one may not entirely agree with them. The statements contribute useful views on a problem of great public interest.

Regarding Data on Ships

It might well become one of the works of the Marine Standards Committee to formulate recommendations regarding the publication of ship data. There is a very confusing medley of dimensions in use today. Lengths of vessels are quoted sometimes by the overall measurement, at other times by the water line measurement and again in a large number of cases between perpendiculars. Tonnages are given sometimes in terms of displacement, but more often in gross register and less frequently in net register, but on top of these spreads the confusion of deadweight tonnages, which sometimes refer to total deadweight carrying capacity and at other times are based on net cargo carrying capacity. Power is perhaps even more confusingly quoted. Shaft horsepower seems pretty definite. Indicated horsepower was a very useful measurement in the days when the reciprocating steam engine was supreme, but today is sometimes used in connection with motor vessels and not always on the same basis. Indicated horsepower means the same thing in both 2-cycle and 4-cycle engines, but there is a difference in the proportion of indicated horsepower applied at the propeller in these two different classes of machinery. Furthermore, indicated horsepower has been used by some shipbuilders not as a measure of the power actually shown by indicators on an oil engine, but to express the power of a reciprocating steam engine that would deliver the same horsepower on the shaft. Even these three items which have been briefly discussed have other terminological variations. There are besides other useful ship dimensions, which will occur to everybody and need not be specifically mentioned, that ought to be on a standard basis. Some sort of order ought to be brought out of this chaos. It would be a good work for the Marine Standards Committee to build up recommendations on the subject to serve as the basis for international discussion and possibly international agreement.

A Useful Magazine

With the continuing progress being made in the stationary type of oil engine there is a lot of technical development that cannot be recorded in this magazine because it does not fall within the marine domain. In the early days of the motorship it was usual to make reference to the Diesel engine installations on land. When the motorship became more widely accepted it became responsible for the development of the marine oil engine along special lines, and this development outran the practice of stationary engines. There ensued a period during which the marine oil engine was quoted as an example of what could be done ashore. Now the stationary oil engine is beginning to match the marine oil engine in point of size and importance. It will interest many marine engineers to read in the February issue of OIL ENGINE POWER that a powerful Diesel plant has been ordered for the Panama Canal, comprising three generating units of 3750 b.hp. each. Another article in the same issue deals with the operation of a Diesel power plant of 3500 kw. capacity during a period of eight years. There is a striking article dealing with the features of the new Fairbanks Morse Diesel engine, not previously described in any magazine. Further developments of the Diesel locomotive are also covered. No marine engineer aspiring to keep acquainted with oil engine development should fail to read the last issue of that very informative magazine.

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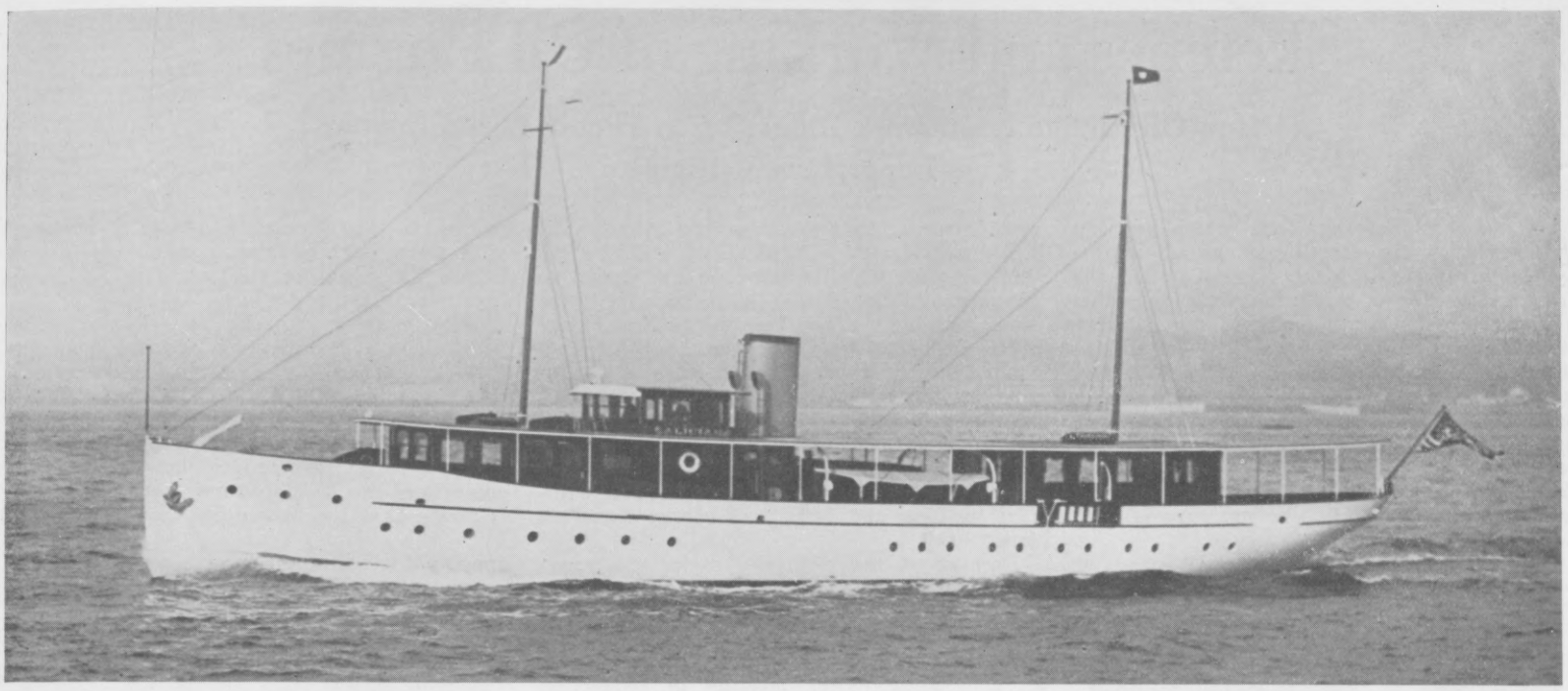


C. W. Wiley's Diesel-driven yacht ALICIAN recently placed in service on Puget Sound

One of the Pacific Coast's Finest Yachts

"Alician," a New Twin-Screw Diesel Driven Pleasure Craft, Makes Her Début in Seattle Waters

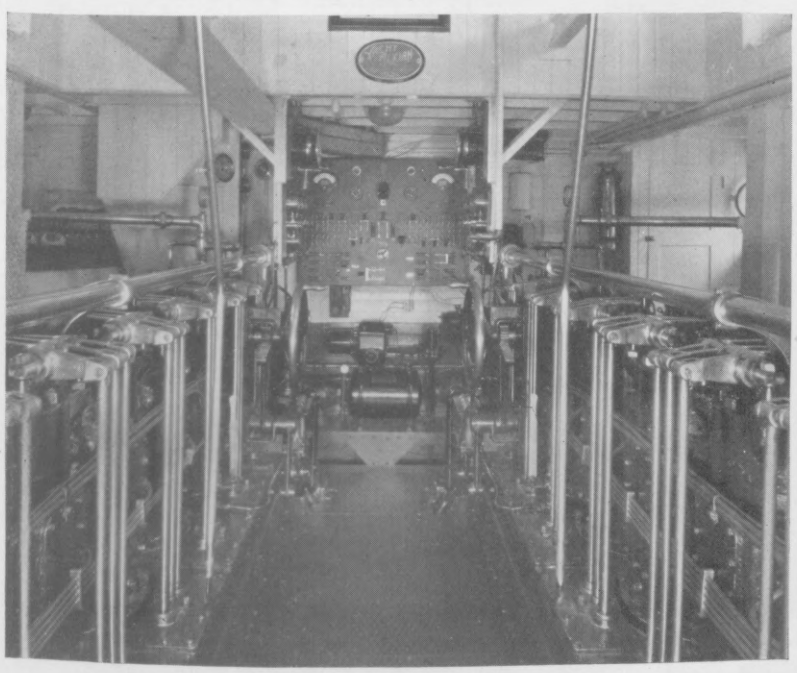
THE various merits of oil-engine power aside from the question of economy have become apparent to yachtsmen on the Pacific Coast, partially due to the rapidly increasing number of oil-engined workboats in western waters.

In the present case the owner—C. W. Wiley of Seattle—of the new twin-screw Diesel yacht ALICIAN had a considerable amount of first-hand experience with the installation and use of oil-engines in vessels of various types, so has had plenty of opportunity to become impressed by this form of drive. His new yacht has just been placed in service and is commanding considerable attention on the Pacific Coast. She is 116 ft. long and is powered with twin 6-cylinder 150 b.hp. Winton engines.

Sleeping accommodations are provided for eight persons in addition to the crew of six men. Fuel tanks of 4,200 gal. capacity give her a cruising radius of several thousand miles at 12 knots speed. No attempt has been made to crowd accommodations for a large party into the boat, the aim having been rather to make every provision for the complete comfort of a moderate sized party on a long trip. To this end the saloons and staterooms are large and roomy, with plenty of opportunity to move about. The available space has been used economically, but not parsimoniously. Similarly, the furnishings everywhere are planned to be tasteful and adequate to every need of the owner and his guests without going to the extreme in elaborateness and extravagance.

The ALICIAN was built at the Seattle yards of the Todd Shipyards Corporation, and much of her equipment, such as tanks, stack, etc., was supplied by the same concern. The propelling engines have a bore of 7½ in. by 11 in. stroke and turn at 450 r.p.m. driving Hyde 44 in. diameter by 44 in. pitch bronze propellers.

The auxiliary and electrical equipment of the ALICIAN is worked out according to the latest practice in marine and electrical equipment. In the engine-room is an auxiliary air-compressor, which furnishes air for starting the main engines. It is operated by a 12 h.p. electric motor deriving its current from a 96 cell storage battery. A 7½ k.w. generator is direct connected to a Winton gasoline engine.



Engine-room of the ALICIAN showing twin Diesel engines

Owner's stateroom on the ALICIAN

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Sketches and Working of Oil Engines*

How Oil Engine Framework Differs From That of Steam Engines —Importance of Rigidity

As the result of the loads which are imposed on the framing of an oil engine by the pressures incident to combustion, the frame structure of such machines takes on a character distinct from that of ordinary steam or gas engine construction. It will be recalled that in an oil engine it is necessary to use compression pressures which initially produce temperatures suitable for the establishment of favorable combustion conditions before any heat begins to be available from the combustion itself. No matter whether compressed air or hydraulic pressure is used

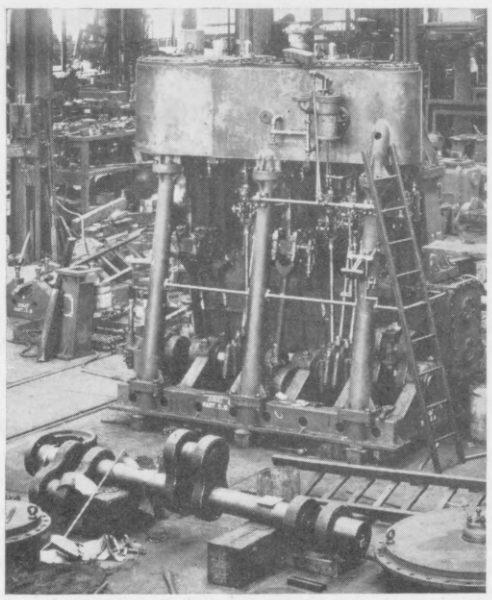

Fig. 24. Open-frame steam engine

for injecting the fuel, total loads about three times as high as those occurring in steam cylinders of equal diameter are regularly encountered.

Avoidance of "Springy" Frames

It is natural to assume that this condition is taken account of simply by using metal sections for oil engine framing which are three times as great as those used for steam work in order that the resulting intensity of stress per square inch of metal shall be the same as that which experience has shown to be safe. Although this is a rational point of view to hold, it has been found that unless other considerations are also taken into account the requirements for successful oil engine practise cannot be fully met. It is necessary not merely to have enough metal for getting a normal intensity of stress per square inch of cross section, but it is imperative also that the distribution of the metal shall minimize yielding and deflections of the structure under load.

Because of the generally easier conditions under which a steam engine operates, elastic deformations of its framing can be greater than they can in oil engines without producing serious consequences. Steam is essentially an easy-going power medium in connection with which relatively large cylinder diameters can be and are employed. They have the effect of opening out the engine framing in a way which has given the steam machine its justified reputation for accessibility. Particularly if a triple expansion engine is taken as an example it is readily apparent how the mounting of the large low-pressure cylinder makes for a wide-open construction. The crank and main bearings are so small compared with the cylinder diameter that they are quite lost beneath its shadow and although the L. P. and the I. P. cylinders are crowded together as closely as the designer can get them, there is generally enough room on the bedplate for the accommodation of not one but two main bearings. In fact two such bearings must often be fitted for no other reason than that otherwise the unsupported span of crankshaft would be too great or the main bearing would become excessively long.

Quite the reverse is true in most oil engine frame constructions. There the effort to get the cylinder centerlines close together can easily succeed because of the considerably smaller cylinder diameters which are used. Because of the higher pressures, however, the loads on the piston rods are the same as or greater than in the steam engine and the decreased cylinder center distance neatly suffices for crank-webs of sufficient thickness and bearings of adequate length. In most oil engine designs the same effort is made as with steam engines to save all possible engine length by mounting the cylinders as closely together as space permits.

With a given cylinder bore the minimum practicable center distance between oil engine cylinder centerlines is generally determined by the outside cylinder head diameter. Either six or eight cylinder head studs are commonly used and the contour which they give the head is what determines how closely the centerlines can be placed together. It is rarely found that the center distance is less than twice the cylinder diameter and an average spacing for many types of engines would be around 2.2, which is of course also equal to the center distance between engine cranks. But in the case of the oil engine 2.2 x D yields a length which, when apportioned among one crank journal, one main bearing and two crank webs just about permits the designer to give these parts the dimensions which the considerations of strength and good wearing demand. Such a thing as shoving in an extra main bearing and having room left over for a coupling and a pair of eccentrics, as is the case with steam engines, does not come into question for oil engine work.

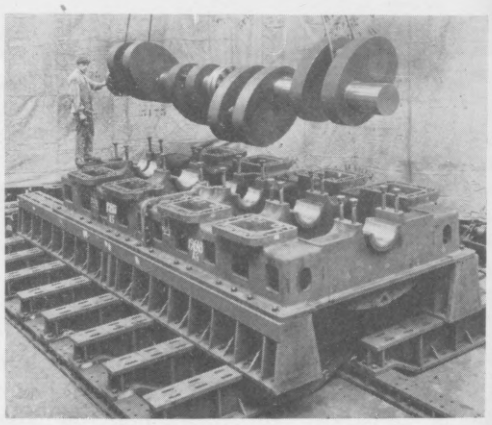

Fig. 26. Diesel engine bedplate, showing closely-spaced column feet and main bearings

Crankshafts for a good many Diesel engines are forged in two sections of three throws each and the latter are joined by a flanged coupling of substantial dimensions. In one of the more modern designs only a one-half main bearing is used on each side of the coupling (Fig. 26) and a considerable economy of space and weight effected. In most engines, however, the coupling practically occupies the same space as one of the crank bearings and spreads out the centerlines of the cylinders next to it so far that there would be almost room enough to get an extra cylinder between them. The contrast merely shows how stringently the space between the cylinders and cranks of an oil engine is economized.

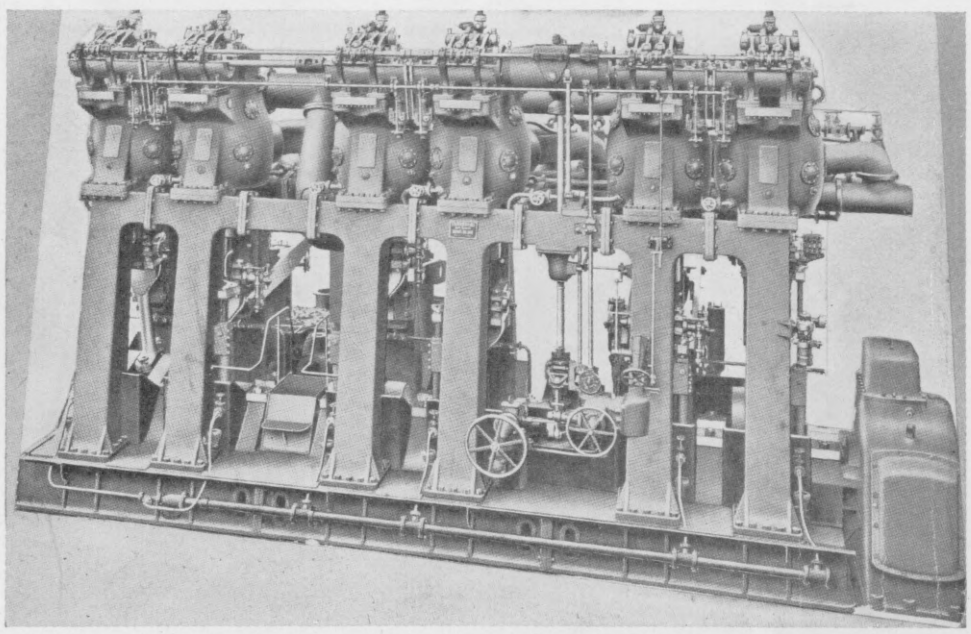

Fig. 25. Open-frame Diesel engine closely patterned after steam engine designs

Closely-Knit Framing

In the early days of oil engine development engineers looked askance at the high degree of compactness which seemed to be almost dictated by inherent oil engine conditions such as high pressures, small cylinder diameters,

*Summary of a course of instruction at the Polytechnic Institute, Brooklyn, N. Y., by Julius Kuttner, B.Sc., Licensed Chief Engineer, Associate Editor of MOTORSHIP. This is the third chapter, the first one having appeared in the January, 1925, issue.

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and massive crankshaft proportions. The wide-open steam engine framing, a typical example of which is shown in Fig. 24, was held as the ideal of simplicity and accessibility.

Characteristic of the steam engine arrangement shown in Fig. 24 is the location of the engine columns in the center planes of the cylinders. Their position also makes them ideal supports for the crosshead guides, which project slightly to both sides of the column trunks. Nothing obstructs access to the crosshead slides and there is nothing to prevent the operating engineers from observing and readily adjusting all the major moving parts.

Cast foot-like projections from the cylinders are flanged to the tops of the columns and corresponding pads on the bedplate receive the lower column ends. While the cylinders act as a sort of bridge spanning the upper column pads, that part of the bedplate which carries the main bearing supports bridges the gap between the lower column feet. The cylinders act as a beam fastened at the ends to the columns and loaded in the center with the steam pressure that reacts on the cylinder covers. The bedplate forms a beam supported at its extremes by the column feet and loaded at the center by the main bearing reactions.

Since, in any engine framing, space must be allowed around the centerline for the reciprocating parts there must always be some kind of a beam structure near the top and near the bottom. The question of importance from the oil engine point of view is that of arranging the beam structure in such a way that deflections or "springing" will be minimized.

If there is no more metal in a beam than is necessary for keeping the stress, that is, the tendency to fracture, within reasonable limits, the beam will also yield appreciably when the load is applied to it. Some of the deflections which are thus produced affect the alignment of the working parts, others do not.

Owing to the large amount of space available in the ordinary steam engine frame the bearing pressures to which the moving parts are subjected may easily be kept at a low figure and the harmful effect of deflections is therefore correspondingly small. In fact there are some who might be a little skeptical about the statement that a massive pedestal of cast iron a foot or two in thickness should yield at all, particularly since the actual micro-measurements which have been made on engines in operation and which prove the existence of such deflections have only recently been made and published.*

Any machinist, however, who has been entrusted with the planing of a large engine bedplate knows that the weight of the casting is alone sufficient to throw it out of line. When the rough piece is first set up on the platen of the planer, every effort is made to block it up in such a way that its weight is evenly distributed on packing pieces. If this is not done the bedplate will be in alignment only so long as it is on the planer and will be true later on only if the strain which existed at the time of machining happens to be duplicated at the final set-up. It is hardly to be expected, therefore that the normal engine loads, which are many times as great as that of any of the individual casting weights, should have no effect in producing deflections.

Some of the engineers who first adapted the Diesel engine to marine work probably did not pay serious attention to deflections, particularly since an engine bolted down to a concrete test block would hardly be apt to show them at all. It was only in the yielding hulls of ships that they began to obtrude themselves. Although shop tests of the earlier Diesel engines intended for marine work netted a wealth of information about many of the engine features, the question of deflections was naturally not included among them and machines which gave brilliant results from non-stop full-power runs lasting many weeks on the test floor were later torn out as total failures from the ships in which they had been installed. Just to what extent factors other than deflections contributed to the early disappointments of the Diesel's first attempts to take to the water is hard to ascertain; on the other hand it is probable that

* "Torsiograms and Vibrograms of Oil Engines," pp. 28-31, MOTORSHIP, January, 1924.

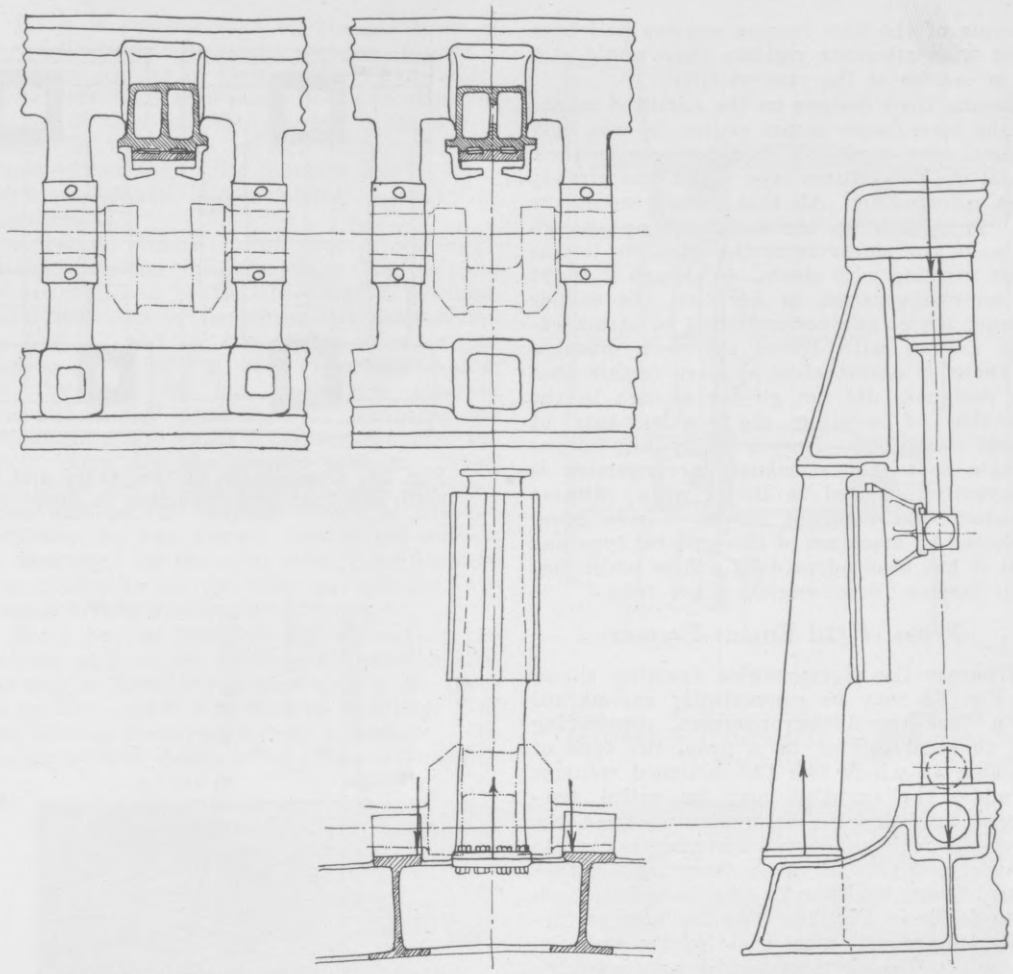

Fig. 27. Yielding of two-column frame due to loads on piston and cylinder head. "Three-legged" frame (left) is better braced

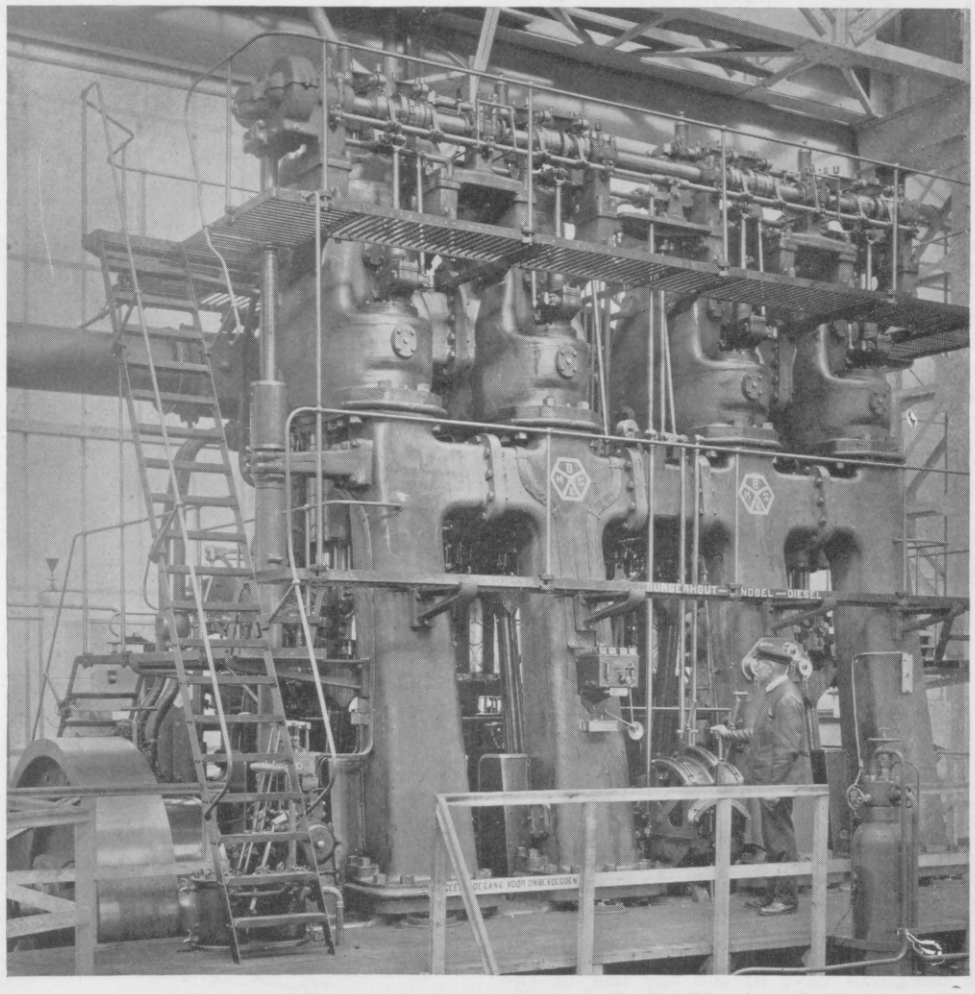

Fig. 28. Steam-type of framing built massive enough for Diesel engine work. Note Siamese connections between cylinders

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if some of the first marine engines had been built with adequate rigidity they would still be in service at the present time.

Basing their designs on the admitted merits of the open-frame steam engine, it was only natural that engineers should choose as their model the two-column type which has already been referred to. All that seemed necessary was to thicken up the metal section enough to make the stress in it the same as it was when working with steam. Although it might be an exaggeration to say that the earlier marine Diesel engines consisted of steam engine frames with Diesel cylinders mounted on them, it nevertheless appears certain that the designers did not go far enough in the direction of meeting the requirements of Diesel operation. Nor is it by any means certain that the two-column arrangement is inherently unsuited to Diesel work; witness the fact that excellent service is now being rendered by machines of this general type and that it has been adopted by a firm which has built marine Diesel engines since 1904.

Types of Oil Engine Frames

Whereas the steam engine framing shown in Fig. 24 may be conveniently ear-marked as a "one-legged" arrangement, considering the forward column as a prop, the type of framing in which two full-sectioned columns straddle the crankpit may be called two-legged in order to distinguish it from the three- and four-legged arrangements to be discussed presently. A characteristic two-legged frame is shown in Fig. 28 and the bedplate for it in Fig. 26. For the sake of concentrating attention on some of the essential points, two- and three-legged arrangements have been sketched in Fig. 27 and again in Fig. 29.

It is hardly necessary to call attention to the fact that in any engine structure every pound of gas-pressure load acting downward on the piston produces an equal and opposite upward load reacting through the cylinder head on all the major engine parts, Fig. 27. A body of gas acts with equal pressure on all the surfaces which confine it, and the entire working of a reciprocating engine, including the production of the twist which turns the shaft, would be unintelligible without a reference to this basic fact. Although in a single-acting engine it is easy for the student to confine his attention to the downward thrust originating on the piston, it is equally essential for him to remember that without the upward reaction communicated to the main bearings from the cylinder head it would be impossible to make the crankshaft revolve. A single force cannot produce rotation: oppositely directed forces not acting along the same line are traceable in the rotation of all bodies whatsoever.

Forces tending to lift the bedplate up come through the column-feet of any single-acting engine, while at the centerline an equal and opposite sum of forces acts downward through the main bearings. Naturally there are also other components of force, such as those due to inertia, the obliquity of the connecting-rod and the side thrust of the crosshead. With the exception of the last named, these are not great enough to influence frame structures in their more important aspects, and it is sufficient for the purposes of this study to note that the two-legged type of frame with columns on the main cylinder centerline furnish an ideal support for the crosshead guides. The transmission of forces through the bedplate is not quite so direct.

Locating the two columns in the center plane means that they must be spaced far enough apart to clear the swing of the crank and of the crankpin box. In order to reduce the width of the bedplate and to save weight some designers have arched the feet of the columns in such a way as to allow the crank cheeks to pass, an expedient which allows the

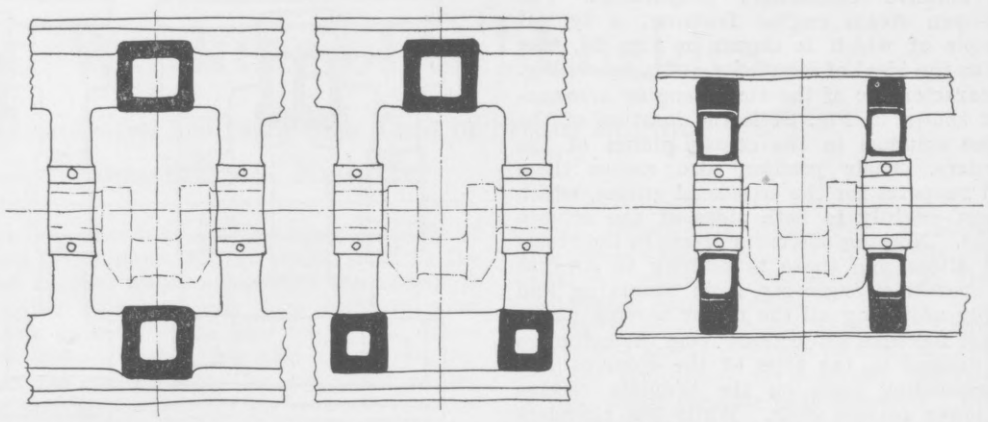

Fig. 29. Comparison of two, three and four-legged framing. Note direct support for main bearings and reduction in bedplate width obtained by four-legged arrangement

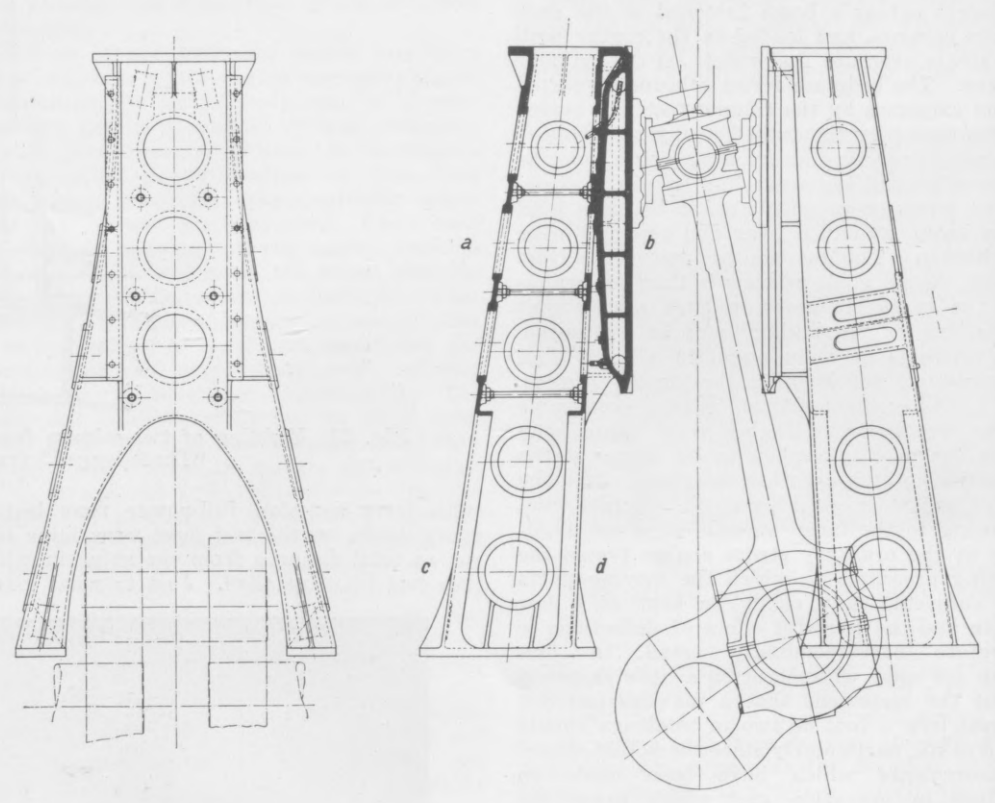

Fig. 30. Two-column frame with column feet arched to clear crank and brought closer in to main bearings

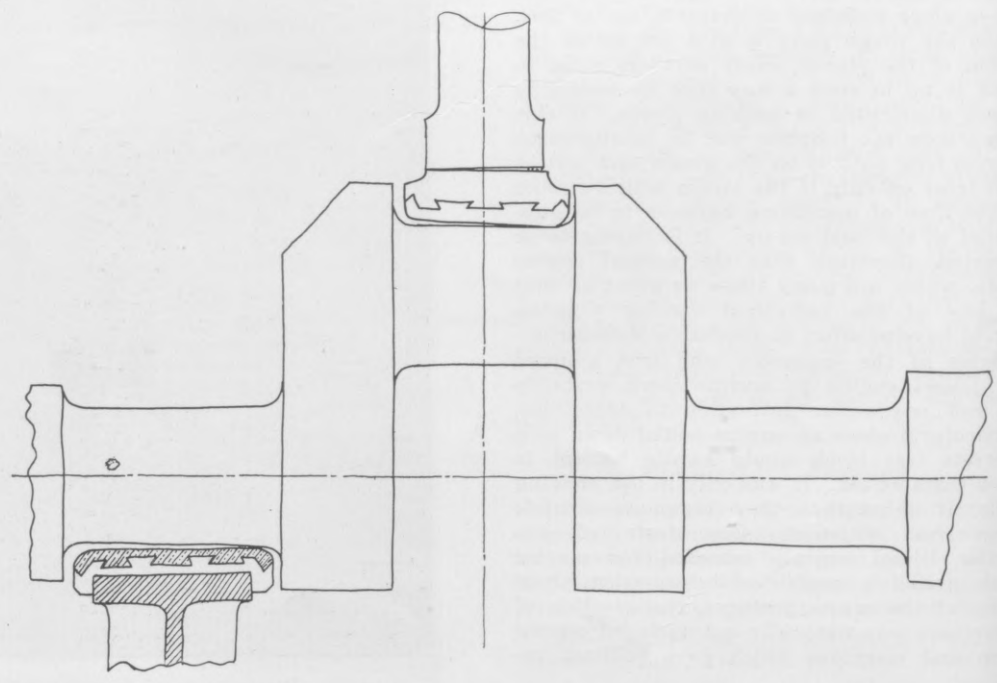

Fig. 31. Deflected main bearing causes journal to bear heavily at one edge

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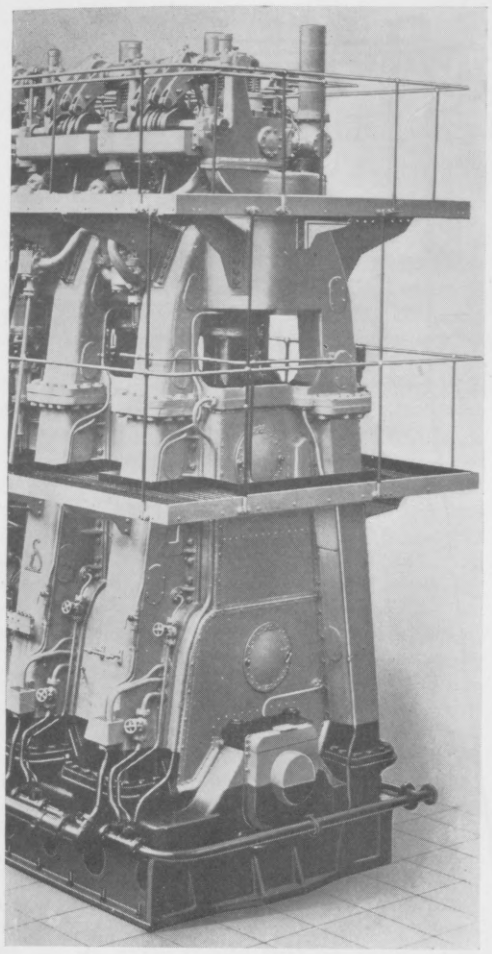

Fig. 32. Three-legged frame of the so-called open type hermetically closed off for pressure lubrication

bolting flanges to be brought closer in toward the main bearings. This is not often done, however, with oil engines, and it is commoner to find undivided column pads similar to those shown in Fig. 26.

Since there is no straight bridge of metal between the column pads of the two-legged type of frame and the main bearings, the bedplate is subject to bending in two directions. Referring to the right hand sketch in Fig. 27, it will be seen that the column and main bearing forces tend to bow the bedplate downward, an effect which has no great consequences as far as the longitudinal alignment of the bearings is concerned.

But the reacting forces are also displaced in a direction along the engine centerline because the main bearings are not in line with the column supports. Bending along the longitudinal centerline is thus encouraged, as is shown in an exaggerated way by the sketch in the center. A bedplate is essentially a grid-like structure consisting of a pair of longitudinal girders connected in the way of the main bearings with transverse girders. Springing the transverse members merely pinches the main bearing shells, whereas the bending of the lengthwise members cants the bearings and has a tendency to make the crankshaft journals ride hard on one edge, as is shown with considerable exaggeration in Fig. 31.

The lubrication of bearings already quite heavily loaded under such a condition is apt to be interfered with. Whereas a thin film of good lubricating oil may stand bearing pressures up to 5000 lb. per sq. in. so long as an effective film area corresponding to such a figure is actually available, the film is quite apt to be ruptured if bearing misalignment throws the entire load on a limited area of the bearing, such as the "corner" shown in the sketch. In that case the true bearing pressure on the oil film is not the quotient of the journal load and the projected area of the bearing but the load divided by a narrow bearing area of very limited extent.

Little is accomplished by the provision of large bearings accurately scraped if only a small percentage of their areas remain active when deflections take place. In apparent recognition of this fact there seems to be a tendency in the more modern designs to use short rather than long journals and to make both crankshafts and bedplates as stocky as possible.

A partial elimination of some of the longitudinal bending moments from the bedplate is accomplished by the three-legged arrangement indicated by the upper left hand sketch in Fig. 27 and by the outside view of the engine shown in Fig. 32. The single column at the back of the engine is retained because of the convenience in mounting the crosshead guide which it affords, while at the front there are two smaller columns. Since the latter are in line with the main bearings, that part of the bedplate which is situated between the two column feet is not subject to bending. On the other side of the bedplate the conditions are the same as with the two-legged arrangement.

Much can be done by way of making the column pads on the bedplate of generous size. As will be seen by reference to Fig. 26, there is so little space left between the pads that the bending there can hardly be serious. The bedplate there shown is for a two-cycle engine

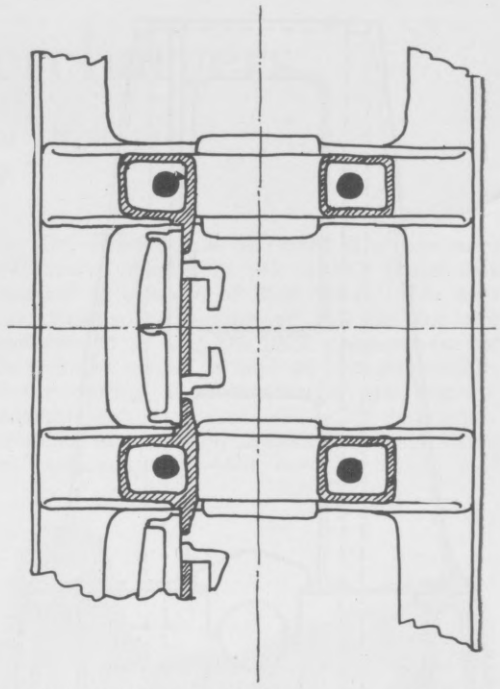

Fig. 35. Sketch showing cross section of four-legged frames with tie-rods close to main bearings

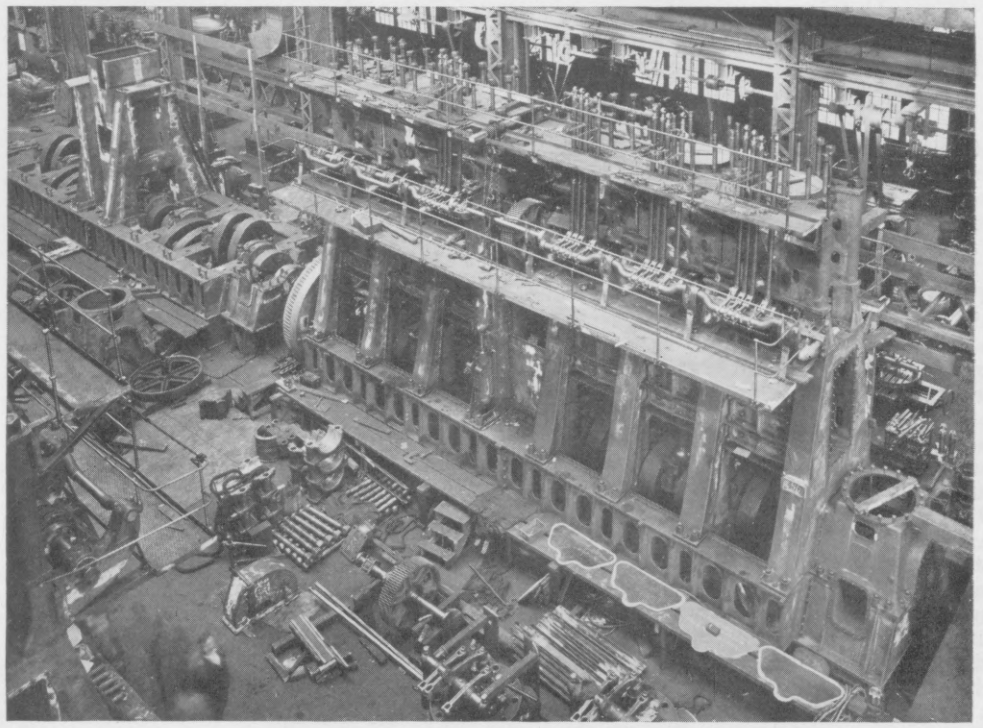

Fig. 33. Diesel engine framework in the course of erection showing fixed part of structure located close to moving parts

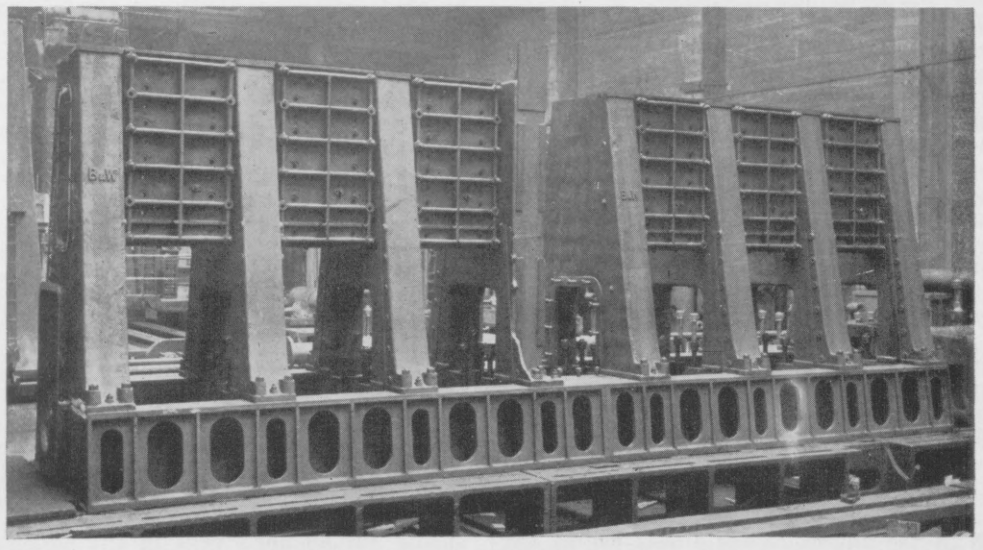

Fig. 34. General view of four-legged framing showing attachment of crosshead guides

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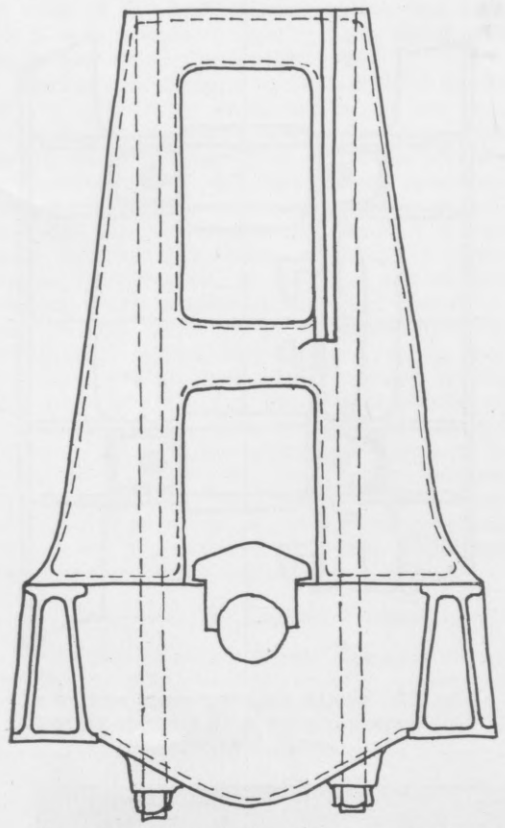


Fig. 36. Span of girder supporting main bearings is kept short

and the shortening of the entire structure due to the use of only four cylinders is in itself a good safeguard against bending deflections.

"Four-Legged" Framing

Placing the engine frames in the way of the main bearings produces the four-legged arrangement sketched at the right hand side of Fig. 29 and photographically illustrated in Fig. 34. Although the front and back columns are here united into pairs of single frames, the fact that they are arched to permit of access to the main bearings results in their bearing on the bedplate at four joint faces.

The bolts shown in Fig. 34 at the feet of the A-frames are not primarily depended upon to transmit the engine loads. This is accomplished by means of stout tie rods running through the cored spaces in the frames and tying the entire engine together from the under side of the bedplate to the tops of the cylinder frames. According to the location of the tie rods as indicated in the sketch, Fig. 36, and the shop assembly view, Fig. 33, the load-transmitting rods are spaced so close to the main bearings that none but the most infinitesimal deflections can occur.

There is one respect in which the use of the A-frames in the planes of the main bearings is not quite as convenient as in the case of the two-legged arrangement, and that is with regard to the mounting of the crosshead guides. If the latter is of the ordinary gib type, it naturally comes on the cylinder centerline and must be supported on some kind of a bridge, as indicated in Figs. 34 and 35. The crosshead proper cannot be reached from the rear of the engine, and access from the front is to some extent restricted by the presence of the two frame contours. As far as taking up the side loads from the crossheads is concerned, that is a matter of no great consequence, because it never amounts to much more than 10 per cent of the direct piston load and can be directly transmitted to the frames, as indicated in Fig. 35.

Accessibility Defined

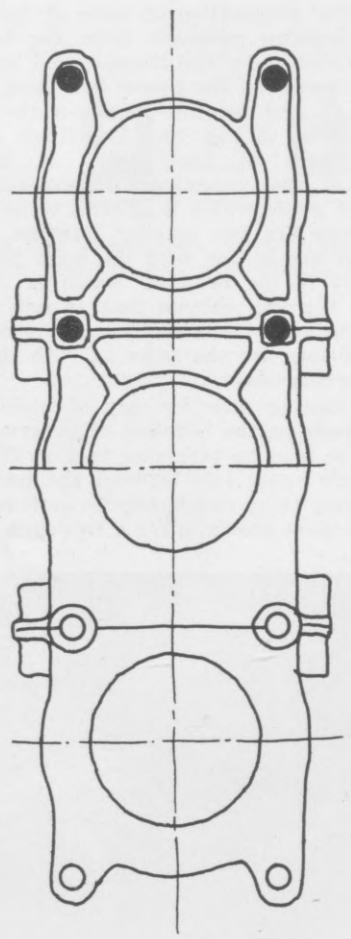
With the advent of the Diesel engine and the structures characteristic of it a certain revision in the meaning of the term "accessibility" appears to have been initiated. Along with the widespread practice of substituting force feed lubrication for the gravity system previously used almost universally on merchant marine steam engines, an almost hermetical sealing-off of most of the working parts was introduced. That rendered the question as to whether this or that frame arrangement is more "accessible" somewhat indefinite in view of the fact that with all those having pressure lubrication the moving parts had to be shut in anyway. The engine shown in Fig. 32 unmistakably has the open type of framing which has formed the basis for oft-repeated assertions concerning the accessibility of steam engine structure; on the other hand, the completeness with which its interior has been sealed off on account of the pressure lubrication would appear to put anything like direct access entirely out of the question.

It would almost seem as though engineers had taken the stand that if the lubrication and construction of the engine parts are right in the first place it is superfluous to watch them in operation. Whether this is the result of an actual conviction or a compromise accepted for securing the advantages of forced oiling may possibly be debated. In either case complete dependence is placed on good workmanship and fitting during manufacture as well as infallible lubrication during operation. It calls to mind the advice of the famous financier who recommended that one should put all one's eggs in one basket and watch that basket. Modern methods of interchangeable manufacture and the use of materials conforming to specifications have as a matter of fact made it superfluous for operating engineers more or less to supplement what are really manufacturing operations with the relatively crude resources available to them while on a watch at sea.

Diesel engines in most cases either run properly or not at all, and, although it would be far from the truth to say that they can get along with incompetent engineers, it seems to be established that the biggest part of the Diesel operating engineer's influence is exerted while the engine is stopped, not while it is under way. If his overhaul work has been good, he has nothing to worry about while on watch; if it has been bad, all the accessibility in the world will help him little toward setting it right while the engine is running.

If this view is valid, then the framing shown in Figs. 33, 34 and 35 is no less accessible than that illustrated in Fig. 32. Since the engine shown in Fig. 25 was intended to run with drip lubrication, splash guards were not used and it may be said to have possessed the same accessibility as the steam engine of Fig. 24. On the other hand, this fact does not seem to have saved a number of marine engines of this make from the scrap pile, while so far as is known none of the machines of the design illustrated in Fig. 34 were ever discarded.

The practice of attaching the framework of a Diesel engine as closely as possible to the main bearings in the manner illustrated by the example under discussion, Fig. 34, comes fairly close to being fundamental.



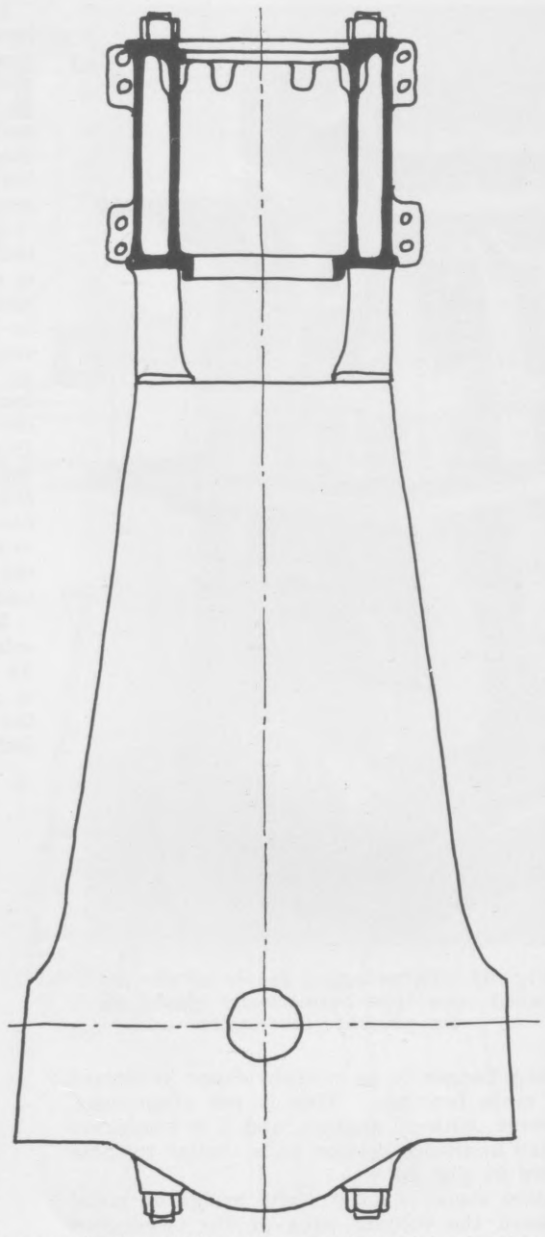


Fig. 37. Framework is tied together at top with beam-like structure formed by bolting cylinder water jackets together

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Exhaust-gas Turbines and Superchargers

Utilization of the Exhaust Gas of Diesel Engines to Drive the Blowers Where Supercharging Is Adopted

IT must be premised that the exhaust gas turbine treated in this article is not a waste heat reclaimer for oil engines of the standard type. It is a development that finds its place only on those engines on which supercharging is utilized, for instance, engines of the type being installed in many German motorships today.

A firm of considerable repute in Europe is responsible for the exhaust gas turbine described below. Though the company is not very widely known in the marine field it has for many years been a leader in the power equipment and electrical equipment fields in Europe, and in one of its departments has for many years been building air blowers of all sizes. During the course of the development of the 2-cycle engine its blowers have been applied to cylinder scavenging. The applications of blowers to supercharging was a natural step. Its blowers are fitted on the Hamburg South American liner MONTE SARMIENTO, described in the last issue of MOTORSHIP, and are to be used also on the sister vessel MONTE OLIVIA. Up to the present, however, most blowers have been electrically driven. (Fig. 1).

The firm in question, Brown Boveri & Co. of Baden, Switzerland, has furnished us with a description of the type of unit it has developed for an exhaust gas turbine drive (Fig. 3). As a sectional drawing of it (Fig. 4) shows, the machine is very simple. A picture of it with the cover removed (Fig. 2) is also given and an interesting comparison is afforded between the pictures of the unit complete (Fig. 3) and the similar picture of an electrically driven blower (Fig. 1).

A single disc mounted on the blower shaft but ordinarily overhung is all that there is inside the turbine casing. The casing is built in such a way that it can expand freely under the influence of heat without any change in its center and without stressing other parts. Owing to the relatively low exhaust temperatures of Diesel engines the materials usually employed for steam turbines can be used for the exhaust gas turbines, especially as the pressures occurring are also as a rule very low.

From the blower shown in the illustrations (Figs. 2, 3 and 4) about 3900 cu. ft. per min. of air can be delivered at a maximum pressure of about 7 lb. per sq. in. It has this capacity at a speed of 8200 r.p.m. For special reasons the nozzles of this turbine are dimensioned so that the back pressure during the exhaust stroke is kept as low as possible. Water cooling is provided for the bearing pedestal next to the turbine. An emergency governor and a hand operated throttle valve are ordinarily provided for regulation, and

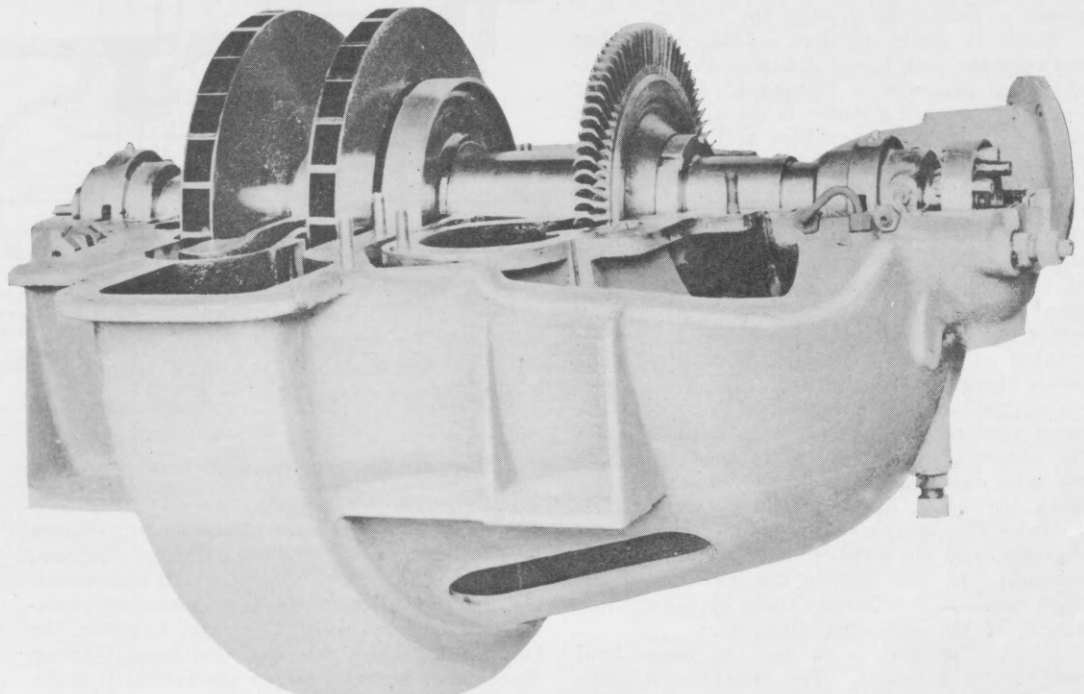

Fig. 2. Supercharging blower driven by exhaust gas turbine, casing removed

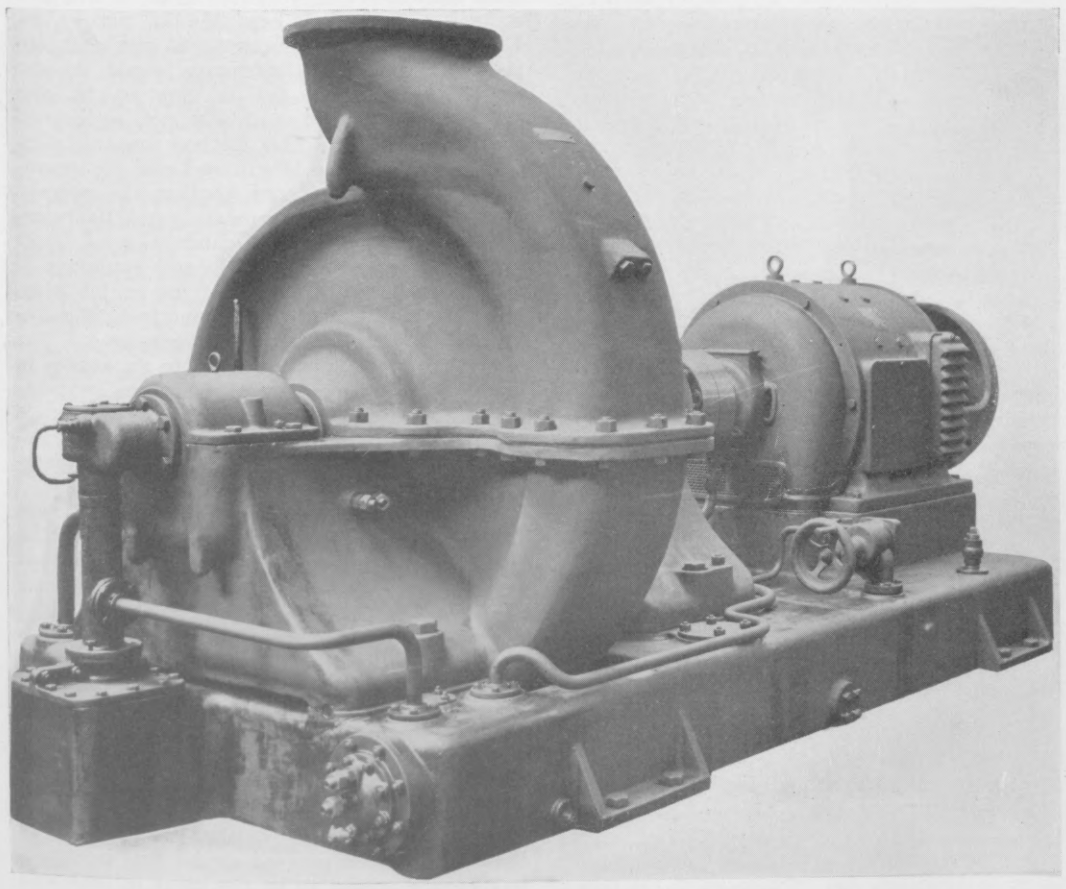

Fig. 1. Electrically driven supercharger of m. s. MONTE SARMIENTO

provision is made for deviating the exhaust gases directly to the muffler when so desired.

Mention was made above of the fact that the use of this exhaust gas turbine is tied up with supercharging. To make this clear a brief résumé must be given of the supercharging process. The purpose of supercharging is, of course, to raise the effective output of any engine. In aeroplane practice supercharging is used at high altitudes to compensate for the loss of power that would result from the diminished atmospheric pressure. Supercharging in aeroplane engines therefore is utilized merely to maintain the power of the engine at its sea level rating. Supercharging in marine oil engines, as introduced by the German firms, involves a different problem. Its purpose is to increase the power of the engine beyond its sea level rating. This can be done in various ways.

Any desired quantity of air can be introduced into a given cylinder volume by a supercharging blower, and theoretically this would permit the mean pressure to be raised to any desired extent. In practice, supercharging is limited not so much by the temperatures of the cylinder walls and valves, but rather by the increase of combustion pressures which would impose too heavy stresses on the working parts. The highest supercharging pressure considered feasible for Diesel engines today is held by Brown Boveri & Co. to be between 6 lb. per sq. in. and 7 lb. per sq. in.

Whether supercharging actually gives an increased engine efficiency under the conditions of ordinary marine service has not yet been fully demonstrated. Theoretically the engine efficiency is improved. Relatively rich

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mixtures are used in oil engines operating on the standard system, and it is partly due to the possibility of employing thinner mixtures that increased efficiency is theoretically obtainable. This is explained by the greater quantity of surplus air in the cylinder permitting more complete combustion. The increased expansion ratio also helps to improve the efficiency.

Further reason why efficiency improves with supercharging is that the no-load losses do not appreciably increase, and consequently the ratio between these losses and the indicated output are diminished. For instance, if one assumes the no-load losses of a standard type of engine to be equal to 25 per cent of the indicated output, then when the power is increased 50 per cent by supercharging the ratio of the no-load losses will be no higher than 17 per cent, deduction being made for the power expended in driving the blower.

There is again further consideration that the cylinder wall losses diminish as the supercharging pressure is increased. The loss of heat to the cooling water is mainly dependent on the absolute temperature and on the temperature differences between the cylinder space and the cooling water. These temperature differences are not altered at all by supercharging, and the absolute pressures are only slightly different when the same composition of combustible mixture and the same expansion ratios are attained. The total cooling water losses remain the same with the increased output, and the specific cooling water losses therefore are reduced.

One method of supercharging is to keep the same compression ratio in the cylinder, and the absolute final pressure is then raised to the same extent as the supercharging pressure. With the same heat content of combustible mixture the temperatures will remain unchanged, and the expansion ratio will remain constant. In this method the increase of the main pressure is effected chiefly by the greater height of the indicated diagram.

Another method is to use the same final compression pressure. The compression space in the cylinder must then be augmented proportionately with the supercharging pressure. The expansion ratio will become smaller. In this method the increase of the mean pressure is obtained by fattening the indicator diagram.

Between these two methods, and incorporating them in different ways, a number of variations can be devised. They are, however, the two principal methods, and the choice between them depends mainly upon the results it is desired to attain.

With low supercharging pressures the first method can very well be used. A supercharging pressure of about 3½ lb. per sq. in. would, for instance, change a final pressure of 455 lb. per sq. in. in a standard engine to a final pressure of about 570 lb. per sq. in. with supercharging.

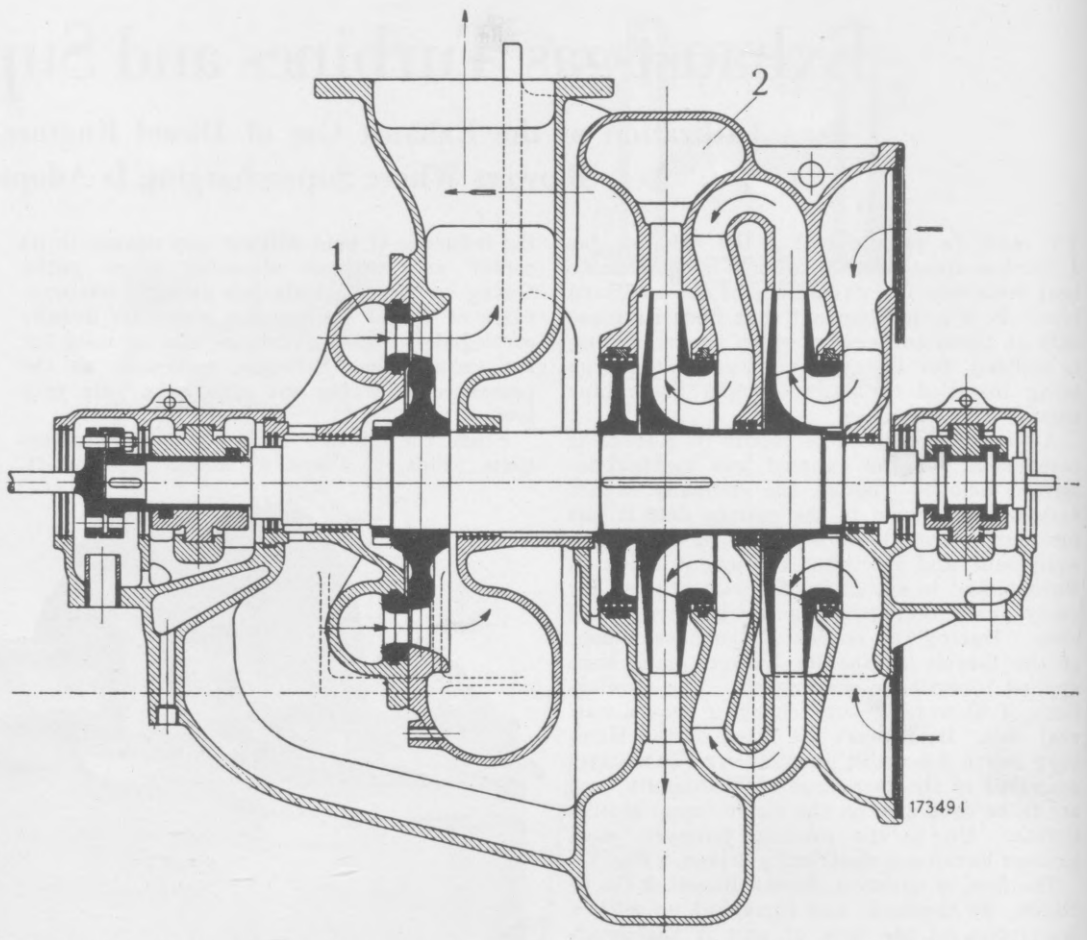

Fig. 4. Section through blower and exhaust gas turbine: 1. exhaust gas turbine; 2. blower

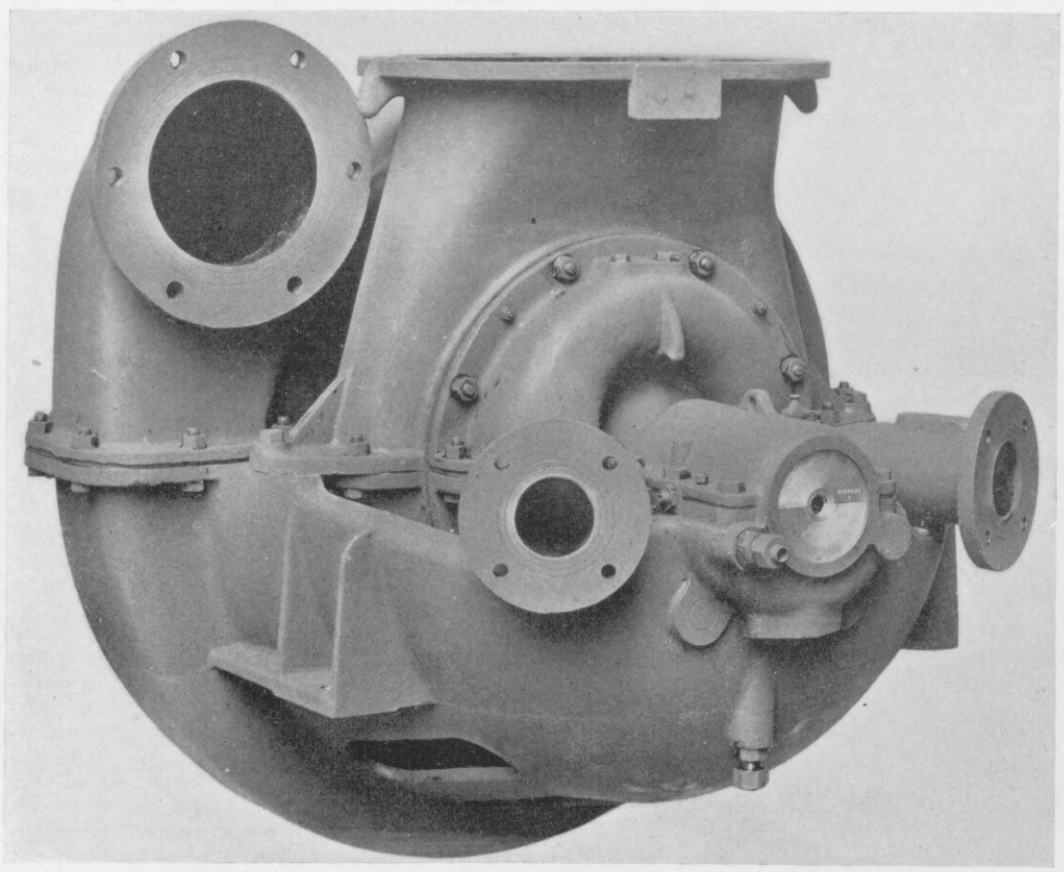

Fig. 3. Exhaust gas turbine and blower set of Fig. 2 complete

When the second method is employed, i.e., when the compression space in the cylinder is increased, thought has to be given to the starting of the engine and to its operation without supercharging. This consideration fixes the compression ratio so that it will give a final pressure of, say, 385 lb. per sq. in. Under these conditions, with 25 per cent supercharge, the final pressure would amount to approximately 485 lb. per sq. in. and the pressure would normally not go beyond 570 lb. per sq. in. This method causes the injection period to be lengthened and the expansion ratio to be lowered, so that the average temperature becomes greater, particularly because also of the relative diminution of losses to the cooling water, as already explained.

To show how the output of the engine is increased by each of these two methods, the diagrams in Fig. 5 have been prepared. The upper group of curves relates to an engine in which the compression ratio has been kept unaltered, while the lower group of curves is for the same machine with an increased compression space, but with the same final pressure, namely 500 lb. per sq. in.

These results were obtained with a 4-cycle Diesel engine of 1000 s.hp. operating at 125 r.p.m. In both cases it has been assumed that the pressure of the compressed air entering the cylinders and mixed there with the slight residue of exhaust gases is the same. It is obviously an assumption that is not quite correct, but it does not introduce any serious error.

These two sets of curves show the indicated output when exhausting directly to the atmosphere and to a constant pressure exhaust gas turbine respectively. The value of the friction losses or no-load losses, plus the loss due to the supply of compressed air for fuel injection, has been assumed to be the same in both sets of curves for all supercharging pressures. The

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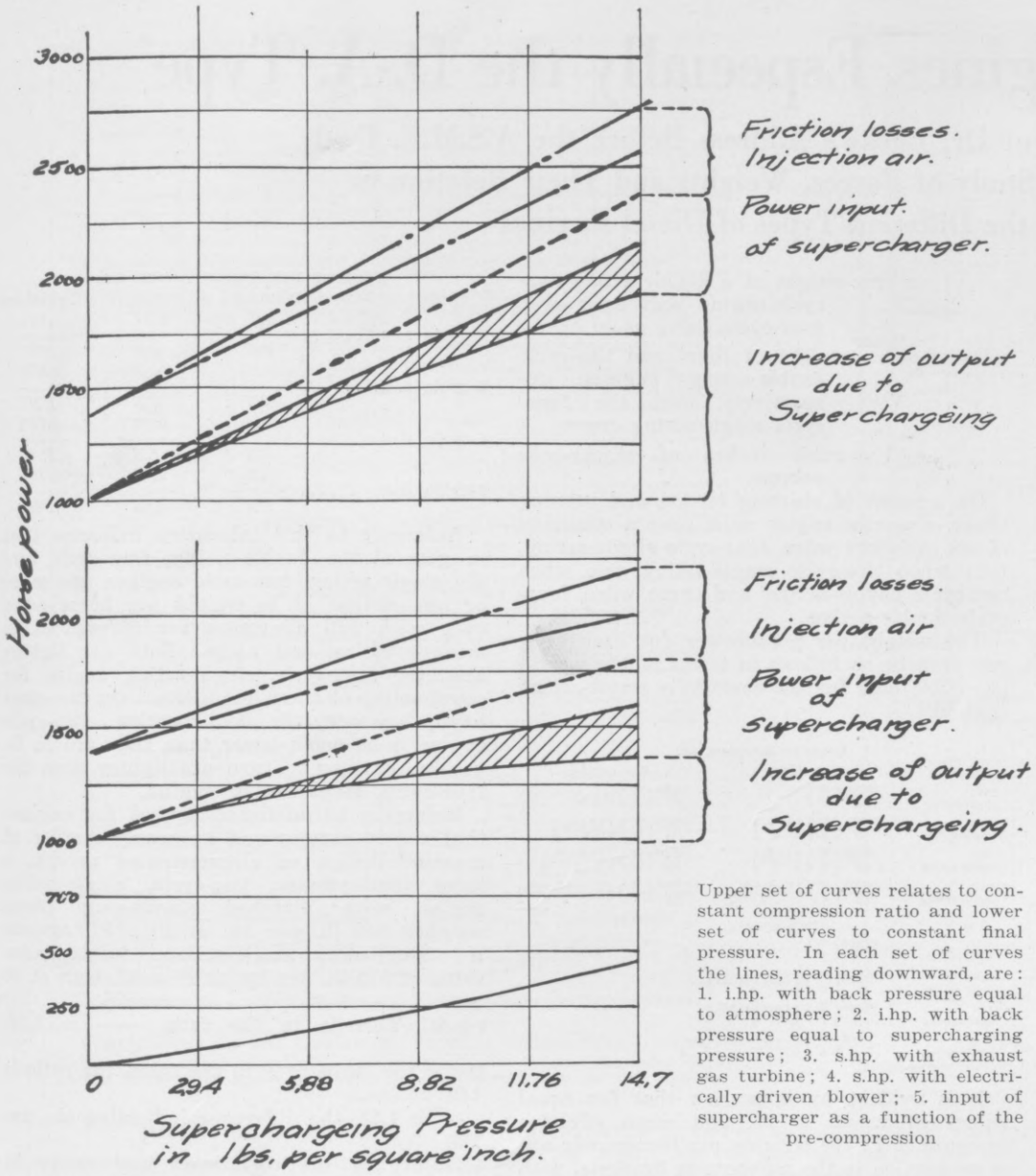

Upper set of curves relates to constant compression ratio and lower set of curves to constant final pressure. In each set of curves the lines, reading downward, are: 1. i.hp. with back pressure equal to atmosphere; 2. i.hp. with back pressure equal to supercharging pressure; 3. s.hp. with exhaust gas turbine; 4. s.hp. with electrically driven blower; 5. input of supercharger as a function of the pre-compression

Fig. 5. Increase of Power with Higher Supercharging Pressures

power necessary for driving the supercharger is given on a separate curve. By subtracting the no-load losses and supercharger load from the indicated power curve one obtains the curve of effective output or brake power. It will be noticed that the gas turbine drive is the more favorable and becomes more economical as the supercharging pressure is raised.

From the two sets of curves it is plain that the increased output depends chiefly on which method of supercharging is employed and between them lies a range of combination.

Having examined the methods of supercharging and the results obtainable thereby, we can revert to the exhaust gas turbine. In standard oil engines there is a considerable amount of heat in the exhaust. In supercharged engines this quantity of heat is greater because of the increase in the mean pressure and of the final pressure of expansion, with a rise of exhaust back pressure.

It is on this result of the supercharging system that the use of the exhaust gas turbine is based. The pressure needed for operating the turbine is not far different from that of the supercharging pressure; that is to say, a back pressure must be created on the exhaust about equal to the supercharging.

There is a choice between two systems of operation for the gas turbine. One can use the variable pressure system which requires the exhaust gases to leave the cylinder at a high velocity. This would require the turbine to be installed close to the cylinders. The constant pressure system seems to have greater advantages. In this system the exhaust gases are throttled sufficiently to give a back pressure, so that they feed to the turbine wheel at almost constant velocity.

A theoretical indicator diagram of a Diesel engine with a constant pressure exhaust turbine is shown in Fig. 6. The supercharging pressure has been assumed equal to 5.7 lb. per sq. in., the swept cylinder volume is taken at 195 cu. ft. and the air temperature is 60 deg. F. Under these conditions the area

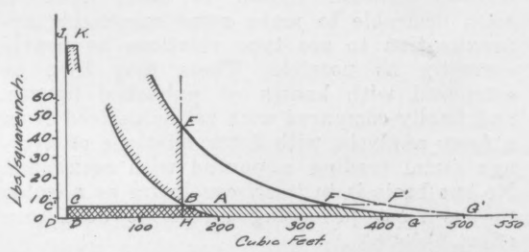

Fig. 6. Theoretical diagram of 4-cycle Diesel with exhaust gas turbine

ADCB shows the work expended by the supercharger in compressing the air and filling the cylinder, while the area *BCJ* is a measure of the work of compression in the cylinder itself. (Part of the diagram is omitted because the full diagram would occupy too much space). *JK* is the line of maximum cylinder pressure and the expansion in the cylinder takes place from *K* to *E*. The point *F* is where the expansion of the exhaust gases in the nozzle commences and, as previously explained, is on the same level as the supercharging line *BC*. The end of the expansion in the nozzle is at *G* on the atmospheric line. The work done in the gas turbine is, therefore, the area *CFGD*.

It has been assumed that the area *EBF* is wasted through impacts and eddies of the exhaust gases flowing past the exhaust valves. The work represented by this waste is mainly reconverted into heat and consequently gives rise to a certain increase of temperature of the exhaust gases when they come to rest before the nozzle. This temperature increase causes the volume *CF* to expand to about *F'*.

If it be desired that the energy developed by the turbine should be exactly sufficient to drive the blower, the difference between the areas *CBAD* and *CFGD* or *CF'GD'* respectively gives the losses that can be allowed in the turbine and in the supercharger.

It has been found that if the pressure ahead of the gas turbine is equal to the delivery pressure from the blower this difference in area is sufficient to cover the transformation losses of an exhaust gas driven blower set having a fairly good efficiency.

The area *BHDC* represents the work expended by the Diesel engine in getting rid of the exhaust gases. This work is, however, returned during the next stroke by the supercharger in filling the cylinder. The amount of exhaust back pressure and consequently the expansion ratio can be fairly well chosen according to the output required from the turbine gas engine. One must remember, however, that work is obtained more efficiently in the Diesel engine than in the exhaust gas turbine, so that the latter should preferably be dimensioned only for providing the output necessary for the supercharging blower.

A comparison between the different methods available for driving supercharging blowers is given in Fig. 7. The values in this diagram have been computed on the basis of a 4-cycle Diesel engine of 1000 s.hp. at 125 r.p.m., with a constant compression ratio in the cylinders for all supercharging pressures up to 14¼ lb. per sq. in., which, of course, corresponds to 100 per cent surplus air. The final compression pressure without supercharging amounts to 500 lb. per sq. in., but is correspondingly altered as the supercharging pressure is changed. The four curves relate to the same blower, driven respectively by each of the following methods: 1, gear drive from the main engines; 2, electric drive; 3, steam turbine drive with steam generated from an exhaust gas boiler; 4, exhaust gas turbine of the constant pressure type with initial pressure equal to supercharging pressure.

In connection with curve 3 it will be noticed that the waste heat for steam raising is sufficient for supercharging pressures up to about 4.25 lb. per sq. in. For higher pressures additional heating is necessary, and in consequence the total fuel consumption increases rapidly.

In conclusion, we may repeat what was stated at the beginning of the article, that this exhaust gas turbine is put forward solely for use with supercharging blowers. It is conceivable that it could be used to drive other auxiliary machinery such as generators, centrifugal pumps, etc., but higher exhaust back pressure necessary for this purpose lowers the output of the Diesel engine and the consequent loss cannot be entirely made good in the gas turbine on account of its lower efficiency.

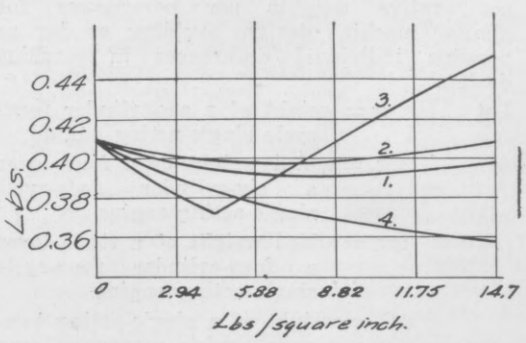

Fig. 7. Fuel consumption of Diesel with varying supercharging pressure and different forms of blower drive (see text)

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Large Oil Engines, Especially the D.-A. Type*

This Section of Dr. Lucke's Address Before the A.S.M.E. Deals With the Study of Forces, Weights and Their Relation to the Different Types of Diesel Engines

AS the maximum pressures in cylinders of all classes of Diesel engines may be taken as equal in any comparative study, it follows that for a given bore and stroke the maximum load in pounds, due to gas pressure, acting to stress the major fixed parts is identical for the two- and four-cycle classes. With negligible or no difference in inertia and centrifugal forces of the major running parts, the same is true for them. Therefore, whatever difference in metal weights there may be in two-cycle as compared with four-cycle engines of equal bore, stroke, and number of cylinders, must be in the minor parts, particularly the scavenging pump weight with connections as compared with the weight of the four-cycle valve gear. Direct comparison of some standard designs of large single-acting two- and four-cycle marine engines of similar framing, of the closed box type, and having crossheads indicates that the weight for each class is about constant when measured in pounds per cubic inch of stroke volume (cylinder area times stroke) for equal numbers of cylinders.

The difference is a measure of the excess weight of the scavenging equipment. Such figures for pounds per cubic inch of stroke volume are independent of cylinder diameter in the range of sizes of interest, but they decrease somewhat with increase in number of cylinders from three to six.

A double-acting engine of the same bore and stroke as a single-acting receives the same head-end loading due to gas pressures, putting the frame in tension, and a loading at the crank end about 10 per cent less because of the rod, putting the frame in compression. Thus, the double-acting cylinder requires no more frame metal than is required for the single-acting to resist these direct loads. The same metal carries its load a large part of the time, just as in a single-acting engine, comparing two-cycle with four-cycle. More metal is, however, necessary to convert a single- to a double-acting cylinder; in the cylinder with its head and a stuffing box, in the piston, and in various other items, including connections. The total is about one and a quarter times the weight per cubic inch for single-acting engines. For similar construction details and materials, that is, similar machine design, the double-acting engines will therefore weigh a little more per cubic inch of stroke volume than the single-acting, and the excess may be greater for two- than for four-cycle, because the cylinder must be longer.

It is desirable, in order to compare the four types, to set down some values for these weights per cubic inch as a basis for judging comparative weights per horsepower for similar machine design, avoiding as far as possible individual differences in machine design.

Let w = weight of a six-cylinder four-cycle single-acting engine

wx_1 = added weight of a four- over a six-cylinder four-cycle single-acting engine

wx_2 = added weight of a three- over a four-cylinder four-cycle single-acting engine

wS_s = weight of a single-acting two-cycle engine scavenger over four-cycle valve gear

wS_d = weight of a double-acting-two-cycle-engine scavenger over four-cycle valve gear

wD_4 and wD_2 = weight of four- and two-cycle double-acting engines, respectively, over the four-cycle single-acting engine

V = cubic inches of stroke-bore volume.

On account of starting torque and turning effort, a marine engine must have a minimum of six cylinders when four-cycle single-acting, four when two-cycle single-acting, two when two-cycle double-acting and three when four-cycle double-acting.

The weight per horsepower for each style will then be as follows in terms of the weight per cubic inch for the four-cycle single-acting engine:

Style	Weight per Horsepower, Lb. Six cylinders	Four cylinders
4-cycle single-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)$	$\frac{792000}{PN}\left(\frac{W}{V}\right)(1+x_1)$
2-cycle single-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+S_s}{2}\right]$	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+S_s}{2}\right]$
4-cycle double-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+D_4}{2}\right]$	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+D_4}{2}\right]$
2-cycle double-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+S_d+D_2}{4}\right]$	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+S_d+D_2}{4}\right]$
Style	Three cylinders	
4-cycle single-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)(1+x_1+x_2)$	
2-cycle single-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+x_2+S_s}{2}\right]$	
4-cycle double-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+x_2+D_4}{2}\right]$	
2-cycle double-acting	$\frac{792000}{PN}\left(\frac{W}{V}\right)\left[\frac{1+x_1+x_2+S_d+D_2}{4}\right]$	

It is evident by inspection that for equal values of r.p.m. $= N$, and mean effective pressure $= P$, the weights per horsepower are in proportion to the numbers in brackets, with that for the six-cylinder, four-cycle, single-acting engine as unity. The numerators in the brackets are the relative weights per cubic inch of stroke volume.

To make a true comparison would require evaluation of the various quantities—which cannot be done precisely because there are no really homologous designs of engines in all classes and styles, nor do various builders adopt the same values of speed and mean pressure for either combustion conditions or wall-temperature limits. It does, however, seem desirable to make some reasonable approximation to see type relations as nearly correctly as possible. These may then be compared with known or published figures, and finally compared with ratios derived from a force analysis, with determinations of average metal loading compared with maximum. No one basis is in itself conclusive as a guide to the most promising directions of future effort.

From some comparisons of weights of a number of three-, four- and six-cylinder engines, it seems that the following are reasonable values, and close enough to the truth for type comparison:

$x_1 = 0.11$; $x_2 = 0.08$; $S_s = 0.12$; $S_d = 0.09$; $D_4 = 0.22$; $D_2 = 0.30$.

$1+x_1 = 1.11$; $1+x_1+x_2 = 1.19$; $\frac{1+S_s}{2} = \frac{1.12}{2} = 0.56$;

$\frac{1+x_1+S_s}{2} = \frac{1.23}{2} = 0.615$; $\frac{1+x_1+x_2+S_s}{2} = \frac{1.31}{2} = 0.655$;

$\frac{1+D_4}{2} = \frac{1.22}{2} = 0.61$; $\frac{1+x_1+D_4}{2} = \frac{1.33}{2} = 0.665$;

$\frac{1+x_1+x_2+D_4}{2} = \frac{1.41}{2} = 0.705$; $\frac{1+S_d+D_2}{4} = \frac{1.20}{4} = 0.35$;

$\frac{1+x_1+S_d+D_2}{4} = \frac{1.50}{4} = 0.375$; $\frac{1+x_1+x_2+S_d+D_2}{4} = \frac{1.58}{4} = 0.395$.

Substituting these values, the weight relations per horsepower are as follows:

Style	Weight per Horsepower, Lb. 6 Cylinders	4 Cylinders	3 Cylinders
4-cycle s.-a.	$\frac{1.0}{PN}$	$\frac{1.11}{PN}$	$\frac{1.19}{PN}$
2-cycle s.-a.	$\frac{0.56}{PN}$	$\frac{0.615}{PN}$	$\frac{0.655}{PN}$
4-cycle d.-a.	$\frac{0.56}{PN}$	$\frac{0.665}{PN}$	$\frac{0.705}{PN}$
2-cycle d.-a.	$\frac{0.35}{PN}$	$\frac{0.375}{PN}$	$\frac{0.395}{PN}$

Reference to this tabulation indicates that in general the double-acting, four-cycle and the single-acting, two-cycle engines are more or less similar as to weight per horsepower even with full allowance for differences in mean pressure and r.p.m. Both are lighter than the four-cycle, single-acting engine for equal values of these quantities. On the other hand, however, the double-acting, two-cycle engine is as much lower than they are in lb. per hp. as they in turn are lighter than the four-cycle, single-acting engine.

Referring to published figures for engines that are in service and as nearly similar in machine design as circumstances permit, a large four-cylinder, two-cycle, single-acting engine with attached scavenging pump weighed 340 lb. per hp. at $P = 57$, against a six-cylinder, single-acting, four-cylinder value of 575 lb. per hp. at $P = 62$, both at 90 r.p.m. This is in the ratio $\frac{575}{340} = 1.69$. Using the value of P in the table, the ratio is $\frac{162}{108} = 1.51$, the difference indicating the uncertainty of the coefficients and range in machine design. A two-cycle, double-acting engine of four cylinders and comparable as to speed gives a weight ratio in comparison with the six-cylinder, four-cycle, single-acting of $\frac{575}{209} = 2.75$ with $P = 64$. The table makes the ratio $\frac{162}{59} = 2.57$, about the same degree of difference as above.

While, therefore, the constants that have been estimated cannot be accepted as having the values assigned, they do seem to be close enough for a general type comparison, though not for any detail work.

As the numerators in the brackets are relative weights per cubic inch of stroke volume, it becomes possible with any specific data to make check calculations by a direct valuation of the several sums of the coefficients over all, and then, by difference, each one in turn.

There are given below some weight figures for a six-cylinder, single-acting, four-cycle engine, 26 in. by 42 in., of crosshead design, with corresponding figures for a Worthington 28-in. by 40-in. double-acting, two-cycle engine, and as an interesting check, figures for a triple-expansion steam engine of comparable size, 26 in. by 43 in. by 74 in., and 48 in. stroke.

Classification	Weights of Engines, Lb. Triple Steam, Excluding Boilers, Etc.	4-Cycle 6-Cyl., Single-Acting	2-Cycle 4-Cyl., Double-Acting
Major fixed parts	172,252	258,200	305,540
Major running parts	58,073	118,330	123,500
Other parts	55,959	186,350	186,960
Total	286,274	562,880	616,000

(*Continued on p.* 211)

*Continued from p. 129, February issue.

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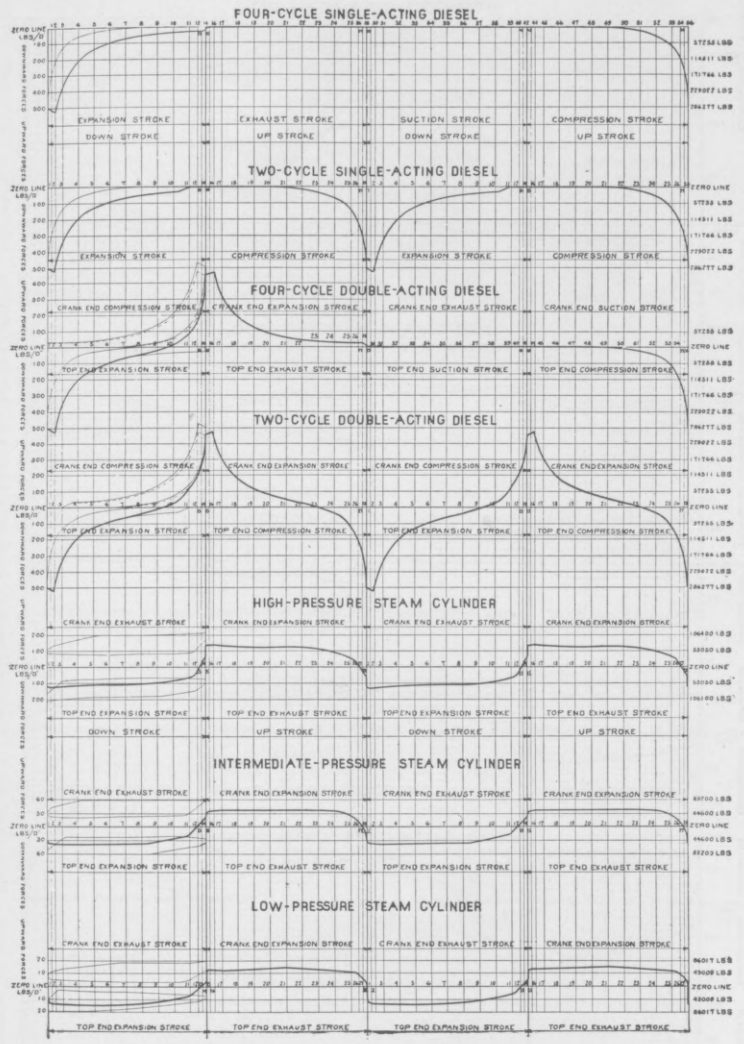

FIG. 11—UPWARD AND DOWNWARD VERTICAL FORCES FRAME LOADS
Upward forces show above, downward forces below, the zero line; forces as plotted apply to piston; frame forces are in the opposite direction

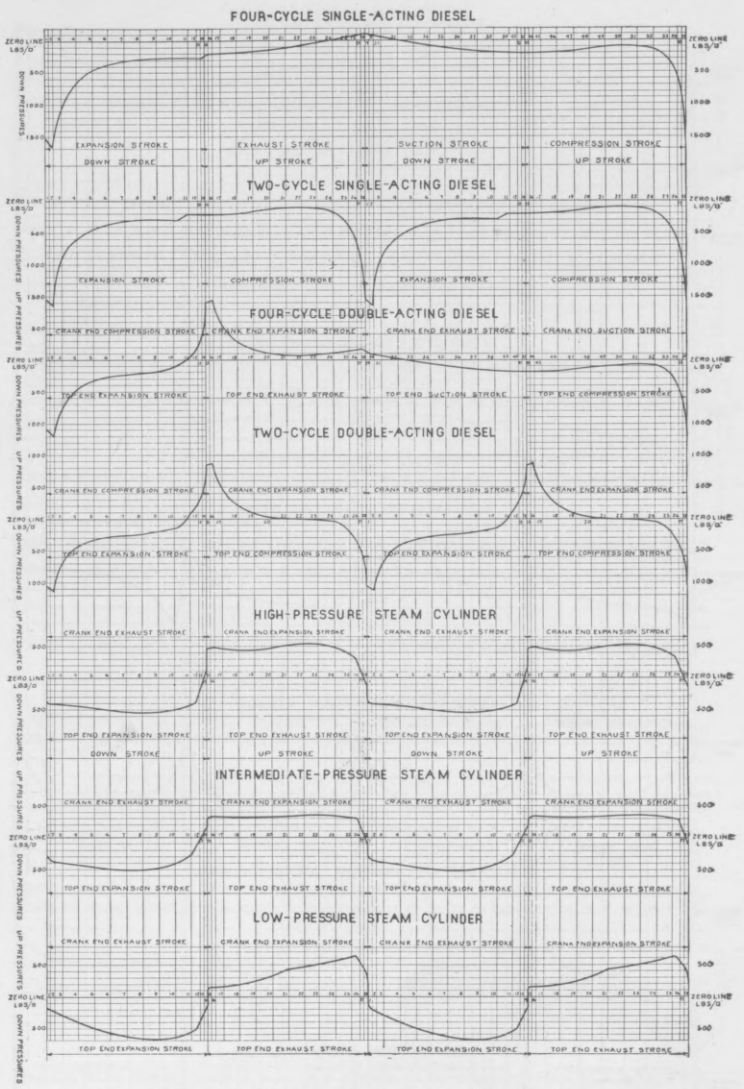

FIG. 12—WRISTPIN BEARING PRESSURES
Pressures below the zero line show downward pressures; above the zero line, upward pressures

(*Continued from p. 206*)

The three designs have stroke volumes of 301,632, 133,812, and 98,560 respectively, which makes the weight per cubic inch 0.95 for steam, 4.20 for four-cycle, single-acting, and 6.25 lb., for two-cycle, double-acting engines, including scavenger. The ratio for the last two is $\frac{6.25}{4.20} = 1.48 = 1+x_1+S_d+D_2$. By addition of the tentatively used coefficients this becomes $1.00+0.11+0.09+0.03 = 1.50$. No one check of this kind is of any particular value, but many cross-checks will in time pretty well establish all these coefficients and their ranges, removing one of the many complexities in the relative characteristics of engines of different types.

One prime variable must always be the unit weights in pounds per cubic inch of stroke volume, and while class totals are given above, it is of considerable interest to be able to compare the principal items in the main parts as follows:

PARTS WEIGHTS, LB.

Major fixed parts	Triple Steam	4-Cycle 6-Cyl., Single-Acting	2-Cycle 4-Cyl., Double-Acting
Bedplate	45,797	64,000	53,230
Frame and housing	39,648	108,000	135,376
Cylinders and liners	66,399	50,000	45,120
Cylinder heads	8,724	31,000	46,000
Main bearings	11,684	15,200	25,820
Total	172,252	258,200	305,546
Major running parts			
Crankshaft	33,088	61,100	44,000
Connecting rods	10,880	30,680	36,360
Crossheads	4,402	9,450	10,960
Piston rods	2,579	9,000	14,840
Pistons	7,124	8,100	17,600
Total	58,073	118,330	123,500

It is only through figures like these that it becomes possible to see what happens when the engine type is changed, especially from single- to double-acting; and even passing from four-cycle to two-cycle with attached scavenging pump, the weight per cubic inch is increased by only 50 per cent while the horsepower is increased to something less than four times for equal cylinders, depending on the other factors discussed, as shown in the tables.

This increase in weight is not the result of overstressing but rather the consequence of the new arrangement, part being due to scavenging pump. The loads on the main parts are shown by the force diagrams and are consistently related, as is clear from the values determined from these diagrams.

Force and pressure diagrams have been prepared over four full strokes for one cylinder of each of the four classes of oil engines, assumed equal in bore and stroke, and for each of the three cylinders of the triple-expansion steam engine, high, intermediate, and low. The first of these diagrams is shown in Fig. 11, and represents to a stroke base the upward and downward forces acting to load the cylinder head, and hence the frame down to main bearings. The regular periodic use of the metal every stroke, typical of the steam engine, is clearly approximated by the double-acting, two-cycle oil engine, but by no other arrangement. In all cases the oil engines have the same pressures over the effective stroke and a low mean value is chosen within range of common use, 94.2 lb. per sq. in. for full-stroke four-cycle and effective-stroke, two-cycle engines, which gives 80 lb. for full-stroke, two-cycle engines. For the steam engine the mean pressures are 98.3 high, 34.5 intermediate, and 11.8 low. These general relations are shown in Table 2, p. 212. From the diagrams the following ratios of maximum to mean loads are found as measures of extent of use of metal:

MAXIMUM AND MEAN FRAME LOADS, LB. PER SQ. IN.

	Upward Forces			Downward Forces		
	Maximum	Mean	Ratio	Maximum	Mean	Ratio
4-cycle, s.-a.	...	...	...	520	42.4	12.26
2-cycle, s.-a.	...	...	...	520	82.6	6.3
4-cycle, d.-a.	470	35.8	13.1	520	40.2	12.9
2-cycle, d.-a.	470	64.6	7.3	520	74.2	7.0
Steam high	136	55.2	2.47	126	51.3	2.46
Steam intermediate	37.4	17.1	2.19	38.8	15.7	2.47
Steam low	15.0	6.5	2.31	14.0	5.6	2.5

The next step in the force analysis is to examine the running parts loading and the bearing pressures. The same cylinder pressures act on the piston and give the piston rod and all connected parts forces equal to those acting on the frame except for the small effect of a piston rod. There are to be evaluated, however, the inertia forces of reciprocating parts and the corresponding centrifugal forces of rotating parts at appropriate points to get correct resultants. The values of these items are given in Table 3, pp. 212 and 213.

Considering the wristpin, the resultant is the combined effect of cylinder pressures, reciprocating inertia, and gravity weight of the moving parts. The reciprocating inertia acts favorably in oil engines to reduce the high initial cylinder pressures and adds to the low terminal or release pressures at the end of the stroke. It is, of course, of no help on

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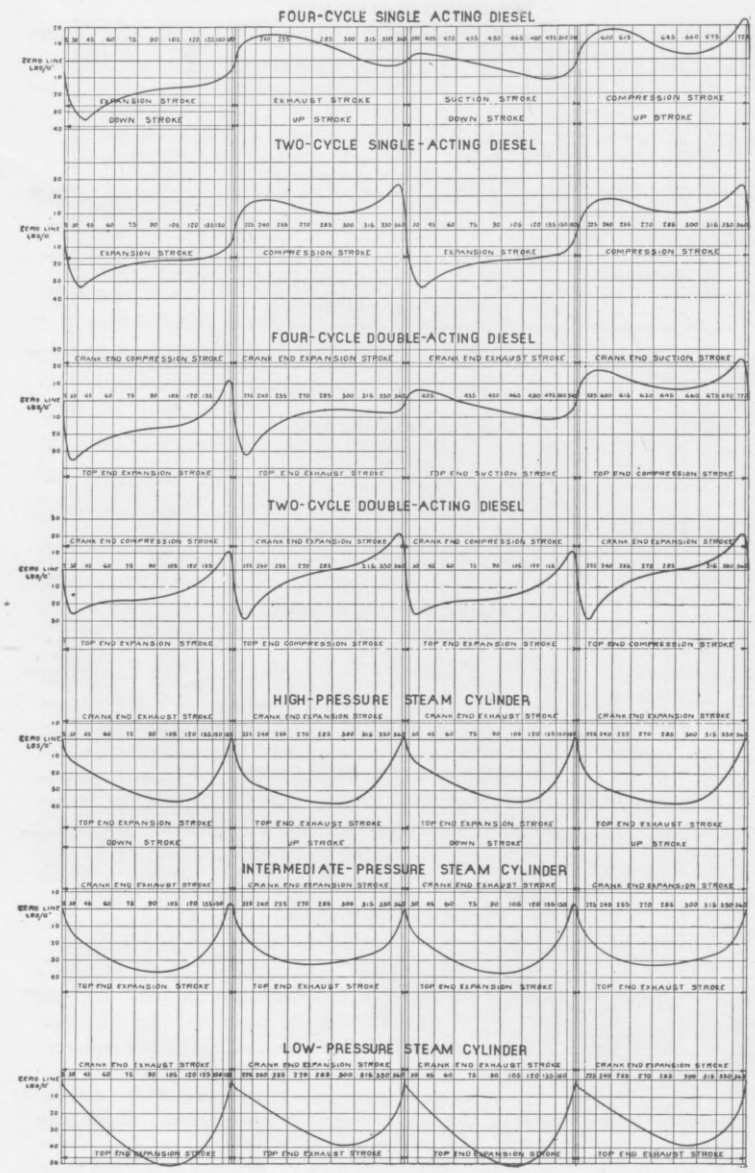

FIG. 13—CROSSHEAD-GUIDE PRESSURES
Pressures below the zero line show pressures on guides; those above the zero line pressures on gibs for ahead running

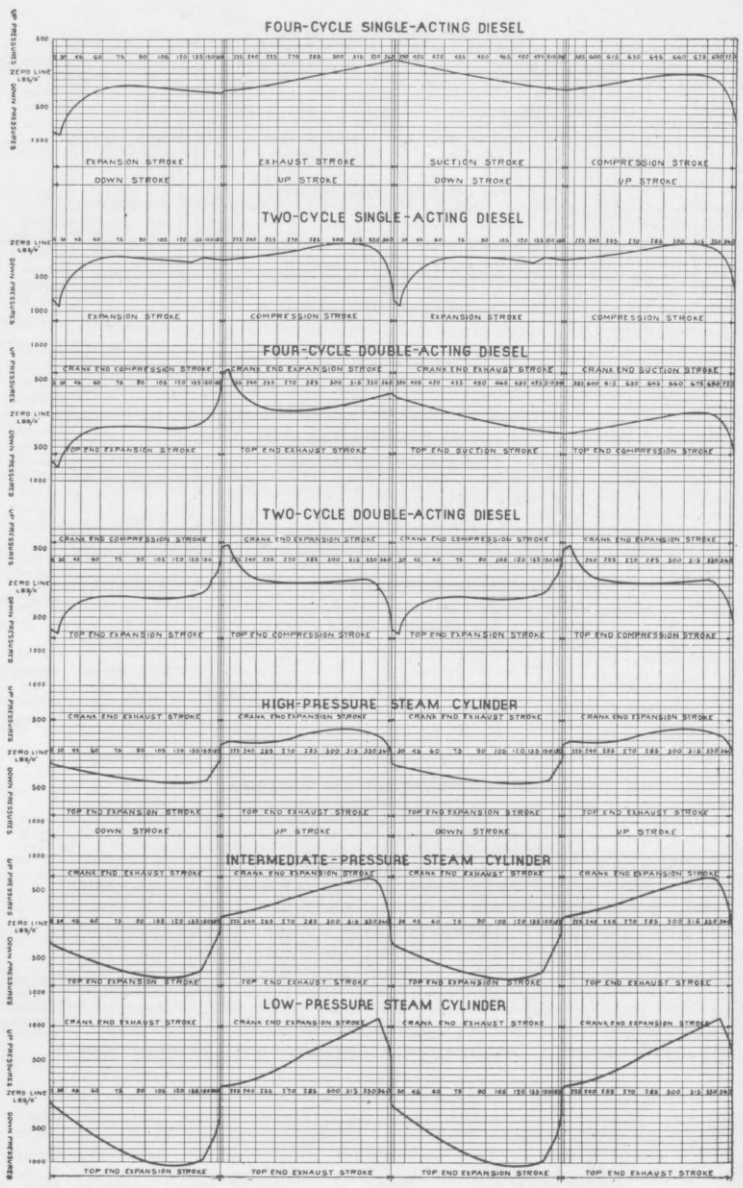

FIG. 14—CRANKPIN BEARING PRESSURES
Upward pressures show above, downward pressures below, the zero line

TABLE 2 FORCE-DIAGRAM DATA

Number of line	Type of engine	Diameter of piston, in.	Area of piston, sq. in.	Stroke, in.	Volume cu. in. cu. ft.	M.i.p. lb. per sq. in.	I.hp. cyl. i.hp.	Ratio i.hp.	Crosshead pin: Diameter, in.	Crosshead pin: Length, in.	Crosshead pin: Area, sq. in.	Piston area: Pin-area ratio
1	4-cycle, single-acting, Diesel engine	27	572.50	40	22900 13.25	94.2	245	100	10	8.5x2	170	3.37
2	2-cycle, single-acting, Diesel engine	27	572.56	40	22900 13.25	80	416	170	10	8.5x2	170	3.37
3	4-cycle, double-acting, Diesel engine	27 27—8.75	572.56 —60.13 512.43	40	22900 13.25 20500 11.85	94.2	245 +219 464	190	10.5	11.375x2	239	2.395
4	2-cycle, double-acting, Diesel engine	27 27—8.75	572.56 —60.13 512.43	40	22900 13.25 20500 11.86	80	416 +372 788	322	10.5	11.375x2	239	2.395
5	High-pressure steam cylinder	26 26—6	530.93 —28.27 502.66	48	25500 14.75 24100 13.92	98.3	569 541 1110	453	7	7.75x2	120.2	4.42
6	Intermediate-pressure steam cylinder	43.5 43.5—6	1486.17 —28.27 1457.90	48	71300 41.2 69800 40.3	34.5	560 550 1110	453	7	7.75x2	120.2	12.34
7	Low-pressure steam cylinder	74 74—6	4300.84 —28.17 4272.57	48	206200 119.5 205000 118.7	11.8	553 551 1104	425	7	7.75x2	120.2	35.8

TABLE 3 INERTIA AND CENTRIFUGAL FORCES

Number of line	Wristpin: Weight of piston and water, lb.	Wristpin: Weight of piston rod, lb.	Crosshead, lb.	Total lb.	Lb. per sq. in. piston area	Crankpin: Inertia coefficient, lb. per sq. in.	Crankpin: ⅔ connecting rod wt., lb.	Crankpin: Total lb.	Lb. per sq. in. piston area	Inertia coefficient, lb. per sq. in. piston area
1	1700	2100	2190	5900	10.3	47.3	3200	9100	15.9	73.2
2	1900	2300	2100	6300	11.0	50.6	3200	9500	16.6	76.3
3	2100	3100	2800	8000	14.0	64.3	4700	12700	22.2	102.3
4	4600	3100	2800	10500	18.35	84.4	4700	15200	26.5	122.
5	1021	860	1467	3348	6.32	8.58	2202	5550	10.46	57.7
6	1751	860	1467	4078	1.746	9.63	2202	6280	4.23	23.4
7	4352	860	1467	6679	1.553	8.58	2202	8881	2.06	11.4

Number of line	Connecting rod, weight, lb.	Total lb.	Lb. per sq. in. piston area	Rotating forces: ⅓ connecting rod weight, lb.	Rotating forces: Lb. per sq. in. piston area	Rotating forces: Centrifugal forces, lb. per sq. in.	Crankpin and webs, lb.	Total on main Bearings, lb.	Lb. per sq. in. piston area	Cent. forces, lb. per sq. in.	Crankshaft: Wt., lb.	Crankshaft: Lb. per sq. in. piston area
1	7100	13000	22.7	3600	6.3	29	3400	7700	13.45	62	10000	17.5
2	7100	13400	23.4	3600	6.3	29	3400	7700	13.45	62	10000	17.5
3	9500	17500	30.6	4800	8.4	38.7	3400	8700	15.2	70	10000	17.5
4	9500	20000	35.0	4800	8.4	38.7	3400	8700	15.2	70	10000	17.5
5	3302	6650	13.25	1700	3.21	17.7	3300	5000	9.43	52.1	9000	16.96
6	3302	7380	4.97	1700	1.15	6.35	3900	5600	3.77	20.8	11400	7.67
7	3302	9981	2.32	1700	0.4	2.21	4500	6200	1.44	7.95	13100	3.05

idle strokes and is harmful rather than beneficial in the steam engine. It is of greatest value in equalizing the resultant forces in the double-acting, two-cycle engine, and higher values for the inertia here used would give better results. This is a direct indication that such an engine may advantageously be run much faster than the speeds here considered. The wristpin bearing-pressure relations in Fig. 12 show clearly the greater regularity of bearing pressures for the two-cycle oil engines and the maximum and mean values compare favorably with the steam engine as tabulated on page 213.

Introducing the angularity of the rod and the areas of the crosshead slides, ahead and astern sides, Fig. 13 results. These pressures are low in comparison with the steam engine, indicating conservative design and perhaps a possibility of later reduction of dimensions, as shown by the tabulation on page 213.

Passing to the crankpin, there is a centrifugal force due to the rod end to be considered in finding the resultant, in addition to the other items, thus making the vertical forces at the crankpin a little different from those at the wristpin. This is shown in Fig. 14 in comparison with Fig. 12, as further modified by relative bearing areas.

Here the peaks due to low reciprocating inertia are still present, but the pressures are low compared with those in the steam engine and would be even better with higher inertia through higher speed. The same situation exists as to main-bearing pressures as shown in Fig. 15 and the general similarity of the two-cycle, double-acting oil engine to the steam engine is very striking. These relations are clearly set forth in the tabulation on page 213.

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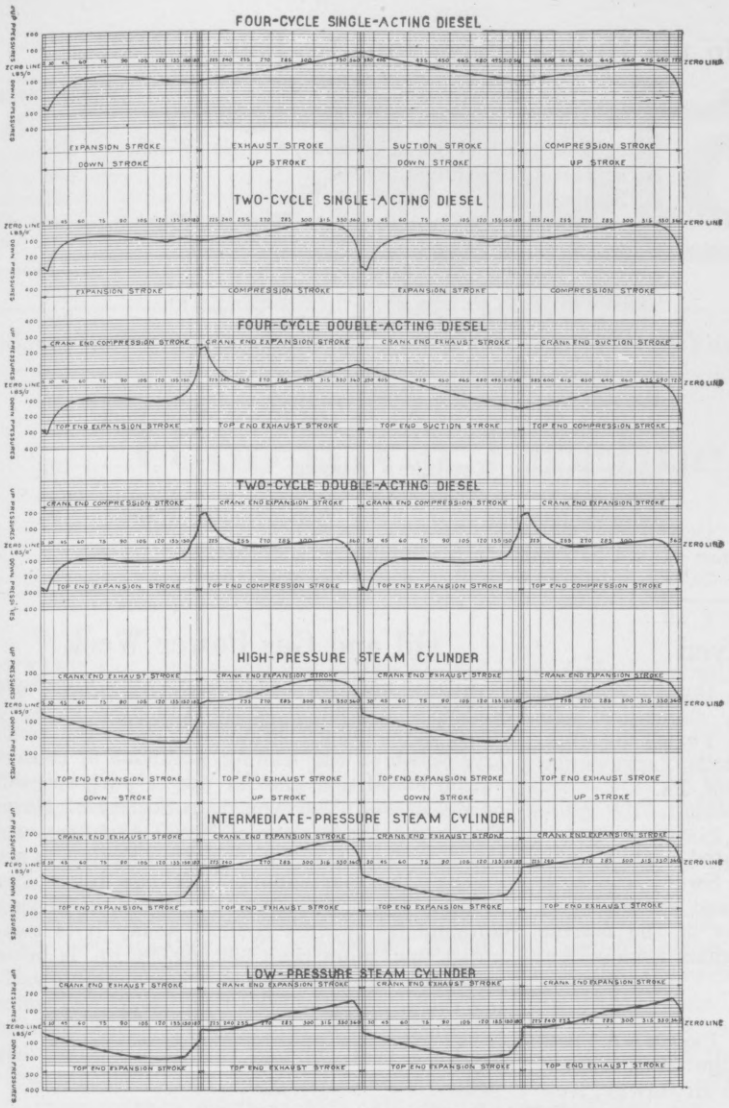

FIG. 15—MAIN BEARING PRESSURES

Upward pressures show above, downward pressures below, the zero line

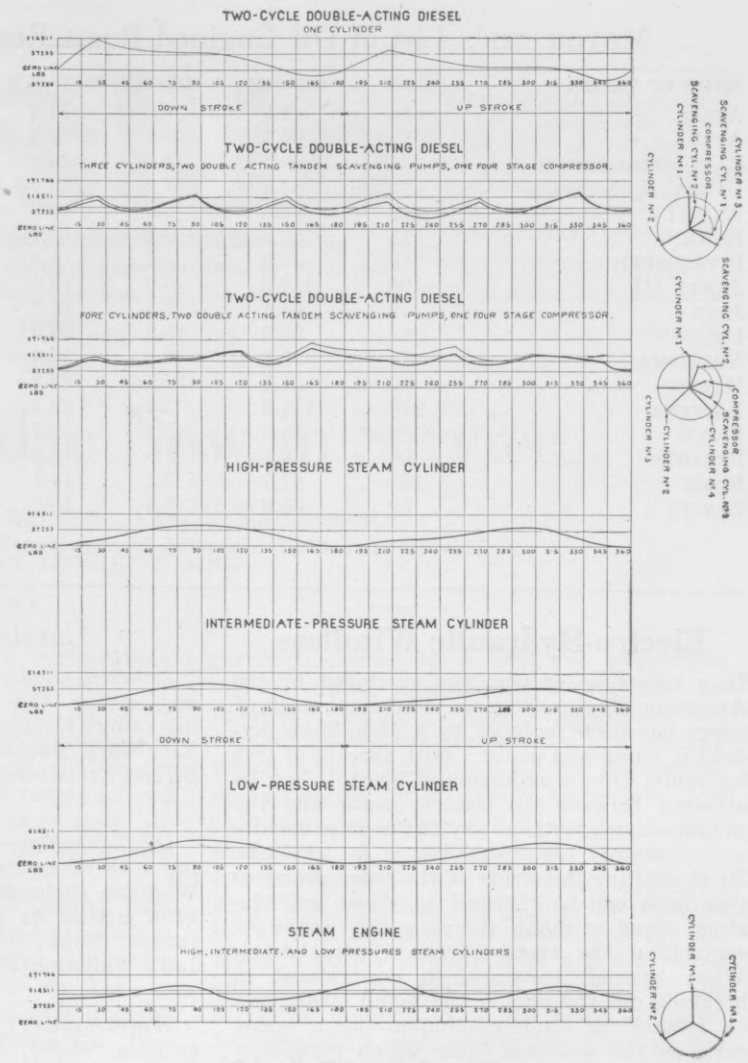

FIG. 16—TURNING EFFORTS

TABLE 3—INERTIA AND CENTRIFUGAL FORCES *(Continued from previous page)*

Number of line	Crankpin Diameter, in.	Crankpin Length, in.	Crankpin Area, sq. in.	Crankpin Ratio	Main bearing Diameter, in.	Main bearing Length, in.	Main bearing Area, sq. in.	Main bearing Ratio	Crosshead guide Width, in.	Crosshead guide Length, in.	Crosshead guide Area, sq. in.	Crosshead gibs Width, in.	Crosshead gibs Length, in.	Crosshead gibs Area, sq. in.
1	18	15	270	2.122	17.5	24x2	840	0.68	22	33.5	737	10.75x2	33.5	452
2	18	15	270	2.122	17.5	24x2	840	0.68	22	33.5	737	10.75x2	33.5	452
3	18	16.75	302	1.9	17.5	21.25x2	744	0.768	21	37	777	6.125x2	37	454
4	18	16.75	302	1.9	17.5	21.25x2	744	0.768	21	37	777	6.125x2	37	454
5	14.5	12.75	185	2.865	14.5 14.5	14.5 14.5	210 210	1.263 1.263	14	26	364	14	26	364
6	14.5	12.75	185	8.02	14.5 14.5	14.5 17.5	210 254	3.54 2.93	14	26	364	14	26	364
7	14.5	12.75	185	23.25	14.5 14.5	17.5 19.5	254 283	8.47 7.60	14	26	364	14	26	364

WRISTPIN BEARING PRESSURES, LB. PER SQ. IN.

	Upward Maximum	Upward Mean	Upward Ratio	Downward Maximum	Downward Mean	Downward Ratio
4-cycle, s.-a.	110	8.9	12.4	1650	236	7.0
2-cycle, s.-a.	...	...	...	1640	321	5.11
4-cycle, d.-a.	940	77.7	12.1	1200	148	8.1
2-cycle, d.-a.	860	86.8	9.9	1140	170	6.7
Steam high	540	219	2.46	540	222	2.43
Steam intermediate	360	131	2.75	500	201	2.48
Steam low	630	177	3.57	690	265	2.6

CROSSHEAD-GUIDE PRESSURES, LB. PER SQ. IN.

	Right Maximum	Right Mean	Right Ratio	Left Maximum	Left Mean	Left Ratio
4-cycle, s.-a.	35	5.87	5.97	24	4.95	4.85
2-cycle, s.-a.	34	9.6	3.54	27	7.38	3.66
4-cycle, d.-a.	35	8.0	4.37	24	3.43	7.0
2-cycle, d.-a.	29	9.8	2.96	22	2.2	10.0
Steam high	38	26.9	1.41	...	...	...
Steam intermediate	37	25.7	1.44	...	...	...
Steam low	51	30	1.7	...	...	...

CRANKPIN AND MAIN-BEARING PRESSURES (VERTICAL) LB. PER SQ. IN.

	Crankpin Up Max.	Crankpin Up Mean	Crankpin Up Ratio	Crankpin Down Max.	Crankpin Down Mean	Crankpin Down Ratio	Main Bearings Up Max.	Main Bearings Up Mean	Main Bearings Up Ratio	Main Bearings Down Max.	Main Bearings Down Mean	Main Bearings Down Ratio
4-cycle, s.-a.	200	17.7	11.3	900	153	5.9	65	5.5	11.8	280	60	4.67
2-cycle, d.-a.	...	...	...	900	211	4.27	...	...	...	285	80	3.56
4-cycle, d.-a.	650	54.8	11.85	800	118	6.78	225	18.8	11.96	300	53.2	5.69
2-cycle, d.-a.	560	50.5	11.1	740	119	6.22	200	13.5	14.85	280	55	5.08
Steam high	370	129	2.88	430	168	2.62	155	45.7	3.39	235	86	2.73
Steam intermed	670	197	3.41	800	293	2.73	150	33.7	4.46	210	78.7	2.67
Steam low	1100	281	3.91	1060	394	2.69	160	36.2	4.42	200	72.7	2.75

The favorable values of bearing pressures so clearly set forth for the oil engines compared with the steam engine of good standard design, is very striking, especially for the two-cycle, double-acting oil engine.

Combining the tangential forces at the crankpin for one crank with the same forces for three and four cranks of the double-acting, two-cycle oil engine gives the turning effort for a three- and a four-cylinder engine when, in finding the resultant, there is also included the negative effort of the scavenging pump and spray-air compressor. This is done in Fig. 16 and also for the three cylinders of the triple-expansion steam engine. Inspection of these turning efforts shows the double-acting oil engine to have a more favorable ratio of positive and negative fluctuations above and below the mean in relation to the steam engine. Evaluation of the areas shows that the maximum single energy increment compared to the mean, which fixes the coefficient of irregularity, has the values given below.

TURNING EFFORT: COEFFICIENTS OF IRREGULARITY.

Three-cylinder d.-a. 2-cycle... $= \frac{0.51}{11.9} = 0.043$

Four-cylinder d.-a. 2-cycle.... $= \frac{0.55}{14.9} = 0.037$

Triple-expansion steam engine $= \frac{0.86}{16.4} = 0.053$

Summarizing this part of the study, it is established that the double-acting, two-cycle arrangement offers the greatest prospect for both weight reduction per horsepower and increase of horsepower per cylinder, unless it should appear to be necessary to reduce mean pressures below values of single-acting engines.

(To be continued)

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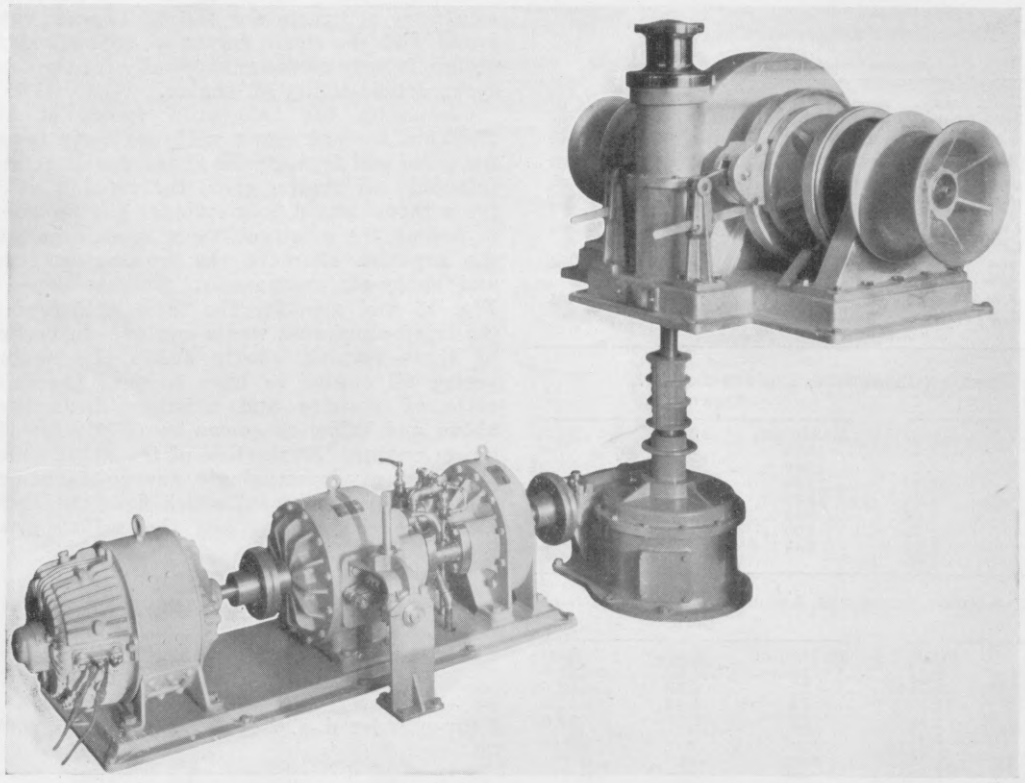
MOTORSHIP's List of Oil Engined Boats Registered in the United States Jan. 1—Feb. 1, 1925

NAME OF VESSEL	GROSS	LENGTH	DEPTH	BREADTH	SERVICE	OWNER AND ADDRESS	HORSEPOWER
ATLAS	14	41.6	4.5	11.6	Towing	W. G. Grant, 1545 Westlake Avenue, Seattle	75
ATTU	55	64.9	7.1	17.0	Fishing	Peter Wold, 6019 Fourth Avenue, Seattle, Wash.	125
CAPT. JOHN SMITH	64	60.5	8.5	24.2	Pass.	A. F. Jester, Smithfield, Va.	60
DESHA H. KIMBALL	75	64.8	5.2	30.0	Pass.	A. Kimball, Arkansas City, Ark.	95
EVA II	22	48.6	5.9	13.6	Fishing	Pietro Marino, 16 Orange Court, Everett, Mass.	45
HYAK	15	30.5	5.2	12.0	Towing	Gilkey Bros. Towing Co., Anacortes, Wash.	75
INTERLAKEN	19	47.0	5.7	11.3	Towing	Shively Towboat Co., Seattle, Wash.	45
JANEY III	190	110.3	5.9	13.6	Ycht.	Consolidated S. B. Co., Morris Heights, N. Y.	400
JEAN	43	65.4	7.4	17.3	Towing	Standard Dredging Co., New York, N. Y.	200
LION	218	95.9	13.8	25.0	Towing	New London Ship & Eng. Co., Groton, Conn.	600
MARITIMA	51	75.2	8.2	16.4	Fishing	Boston Maritime Co., Boston, Mass.	120
MYSTIC	90	81.2	11.6	23.2	Fishing	Boston Maritime Co., Boston, Mass.	90
NIRVANA	236	111.6	7.4	22.3	Ycht.	Rodman Wanamaker, 9th St. and Bway., New York	250
PABCO	16	45.0	5.1	12.7	Towing	J. M. Atthowe, Pier 9, San Francisco, Cal.	75
PARAGON	88	80.1	8.9	18.6	Fishing	Einar Eriksen, Ketchikan, Alaska	165
ROSIE	335	61.7	7.2	16.1	Fishing	John Marino, Boston, Mass.	100
ZENITH	59	65.0	7.9	16.5	Fishing	John K. Iverson, 4231 Latona Ave., Seattle, Wash.	90

This list is carefully compiled from responsible sources but cannot be guaranteed complete or accurate.

Electro-Hydraulic Windlass

In a new type of windlass developed by the American Engineering Co. the usual nest of gears has been replaced by a Hele-Shaw hydraulic pump and motor. With the aid of this hydraulic drive a reduction of 850 to 1 can be obtained between the electric motor and the windlass drum, without any of the noise usually accompanying the operation of a windlass. By controlling the stroke of the pump plungers this ratio can be changed to obtain any desired speed without stopping the motor or controlling the speed of the electric drive. When unusual resistance is encountered in breaking out the anchor the pump stroke can be reduced to its minimum, giving a speed ratio on the windlass drum which permits a much smaller electric motor to be used for the power drive than could otherwise be done.



New style windlass with electro-hydraulic drive

Catalogs Received

Bulletin G—Dealing with thrust bearings and giving the dimensions, capacities and mountings for the Kingsbury Thrust Bearing. Kingsbury Machine Works, 4320 Tackawanna Street, Frankford, Philadelphia, Pa.

Oil Tankships—An illustrated booklet dealing with oil carriers built by Swan, Hunter & Wigham Richardson, Ltd., and containing a brief article on the evolution of the marine tanker. Swan, Hunter & Wigham Richardson, Ltd., Wallsend, England.

Worthington Centrifugal Pumps—A new bulletin W-607, issued by the Worthington Pump & Machinery Corp., 115 Broadway, New York.

Oil and Gas Power Week

Fifteen national technical organizations have united in a plan to devote the week of April 20-25 to meetings throughout the entire country, at which the leading problems of conservation of oil and gas for power purposes will be discussed. The American Society of Naval Engineers, the Society of Naval Architects and the American Society of Marine Designers are among the participants. If one takes into account the horsepower employed in driving automobiles in this country today it is computed that more than 300,000,000 hp. is being generated from oil and gas, completely dwarfing the power developed from steam and water combined. It is to bring this point home to everyone that Oil and Gas Power Week has been planned.

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